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EVALUATION OF DRILLED-BALL BEARINGS AT DN VALUES TO THREE MILLION

I — Variable Oil Flow Tests

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Prepared by
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16. Abstract

Two 125-mm-bore solid-ball bearings and two similar drilled-ball bearings were operated at speeds up to 24 000 rpm (3.0 million DN) with a 13 000 newton (3000 lb) thrust load. The oil flow rate was varied from 45×10^{-3} to 121×10^{-3} kilograms per second (6 to 16 lb/min). The solid-ball bearings operated satisfactorily over the entire range of conditions. The drilled-ball bearing experienced cage rub with marginal lubrication at 45×10^{-3} kilograms per second (6 lb/min). The drilled-ball bearing generally ran cooler than the solid-ball bearings.

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1. Bearings -- Testing

in pts page 2

7Ap 72

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (Cont'd)

LIST OF ILLUSTRATIONS (Cont'd)

LIST OF TABLES

EVALUATION OF DRILLED-BALL BEARINGS AT DN VALUES TO THREE MILLION

I - VARIABLE OIL FLOW TESTS

INTRODUCTION

A recognized need exists in the aircraft gas turbine industry for rolling element bearings having significantly increased speed capability. The successful development of projected engines depends, in part, upon the development of bearings having nearly twice the speed capabilities of those now in use. Current production engines operate in the range of 1.5 to 1.9 million DN (bearing bore in millimeters X rpm), while engines now under development operate at DN values from 2.0 to 2.5 million DN. Engines presently in the conceptual stage will require mainshaft bearing speed capability of 3.0 million DN or greater.

One approach to higher bearing speeds is to reduce the mass of the rolling elements by various hollowing techniques. This reduces the centrifugal load these elements apply to their outer races and may be expected to lead to increased fatigue life at high speeds. Considerable success has been had with hollow roller bearings, and the related "drilled-ball" concept has shown distinct promise in preliminary testing of ball-thrust bearings to speeds of 3.0 million DN.

The experience accumulated by NASA-Lewis Research Center and by Pratt & Whitney Aircraft under Contract NAS3-13491 showed that drilled-ball bearings operate with somewhat lower outer-race temperatures than equivalent solid-ball bearings, at the same conditions of load, speed, oil flow , and oil supply temperature. At the conclusion of Contract NAS3-13491, it was apparent that further exploration and development of drilled-ball bearings was warranted to more fully define their operational characteristics and limits. As a result, the present program was initiated under Contract NAS3-14417 to obtain additional performance data with drilled-ball bearings over an expanded range of test operating conditions, using equivalent solid-ball bearings as a baseline reference. The necessary baseline and drilled-ball bearings were procured and initially tested under Contract NAS3-13491 and required only cage replating and balancing to be suitable for further testing.

This program consists of four distinct tasks, outlined as follows:

Task I	Bearing refurbishment and calibration of the test rig's oil transfer scoops
Task II	Comparison of solid-ball and drilled-ball bearing behavior over a range of oil supply flows
Task III	Comparison of solid-ball and drilled-ball bearing responses to skidding conditions
Task IV	Investigation of drilled-ball bearing durability in cyclic operation

The results of Task II were to be the basis for selecting a bearing oil flow rate for subsequent Task III and Task IV testing. Measurements of outer-race temperature, oil-outlet temperature, and cage speed were to provide the basis for evaluating bearing behavior at the selected test conditions. Thermal stabilization was required at each test point in Task II.

This report describes the baseline and drilled-ball bearing configurations, discusses test equipment and techniques, and summarizes the test results obtained in Task I and Task II. A subsequent report will present the results of the skid-mapping tests (Task III) and the cyclic endurance tests (Task IV). Further assessment of the drilled-ball concept will be made in that report.

SUMMARY

The bearings tested under Contract NAS3-13491 were made suitable for further testing under the present contract by simply replating and balancing the bearing cages. Calibration of the test rig's two oil-transfer scoops in Task I revealed that their performances were identical with a scoop nominal efficiency of 76% ±3% at most combinations of shaft speed and oil supply flow.

Two solid-ball bearings operated successfully to 3.0 million DN in Task II at bearing lubricant flows ranging from 121 x 10^{-3} kilograms per second (16 lbs/min) to 45.4 x 10^{-3} kilograms per second (6 lbs/min). Two drilled-ball bearings performed in a similar manner except at the minimum oil flow of 45.4 x 10^{-3} kilograms per second (6 lbs/min) per bearing at 3.0 million DN. Testing was terminated after 50 minutes at this final test point because the outer-race temperature of one bearing suddenly exceeded the maximum limit of 490.93°K (425°F) established for this program. Subsequent inspection revealed that a cage rub had occurred on both shoulder lands of that bearing under marginal lubrication conditions.

Prior to the cage rub, both drilled-ball bearings operated satisfactorily at all speed levels and lubricant flow rates. Their outer-race and oil-outlet temperatures generally were lower than those experienced with the baseline bearings except for the temperatures of one solid-ball bearing which were slightly lower than the two drilled-ball bearings at 2.8 and 3.0 million DN for most lubricant flow rates. Thermal equilibrium was attained at all shaft speeds and bearing lubricant flow rates, usually within 20 minutes after setting the test point. At 3.0 million DN, the minimum outer-race and oil-outlet temperatures for all bearings were produced at the maximum lubricant flow rate of 121 x 10^{-3} kilograms per second (16 lbs/min).

There was no evidence of ball skidding in these tests. The silver plating in the cage bore of the undamaged drilled-ball bearing was in excellent condition and showed less wear than the cages of the solid-ball bearings. The silver plating was slightly blistered on some of the ball retaining pins and was cracked and separated from the pin in a few instances. Closer examination revealed that a poor bond existed between the base material of all pins and the silver plating. All cage pins were tightly attached to the cage rails, and ball contact marks on the pins and in the cage ball pockets were light.

The condition of the rubbed cage was identical to the undamaged cage except for the bore area. Contact with both inner rings had penetrated the silver plating on the cage bore along both rails. Material had been removed in an arc sector of about 3.14 radians (180 degrees) to depths ranging from 0.05 to 0.76 millimeters (2 to 30 mils). Material had also been removed from the land surface of the two inner-rings.

There were no damaged balls in either drilled-ball bearing. The balls did contain some light orbital lines and random surface-microscratches. Various amounts of oil-sludge were coated to the bore surface of thirteen balls from the rubbed bearing.

Specific results obtained during Task I and Task II testing are summarized as follows:

- The test rig's two transfer scoops have a nominal efficiency of 76% ± 3% at most combinations of shaft speed and oil supply flow.

- The drilled-ball bearings generally operated over the range of lubricant supply flows with somewhat lower outer-race and oil-outlet temperatures than experienced with the solid-ball bearings.

- Marginal lubrication of the bearing inner-ring land surfaces occurs at the minimum lubricant supply flow of 45.4×10^{-3} kilograms per second (6 lbs/min) because of insufficient oil flow through the limited number of shoulder oil passages.

- Drilled-ball motion is relatively stable at a constant thrust load and high DN levels as demonstrated by very light ball contact-marks on the pin silver-plating.

- At 3.0 million DN, the minimum outer-race and oil-outlet temperatures for the drilled-ball and solid-ball bearings were produced at the maximum bearing lubricant flow of 121×10^{-3} kilograms per second (16 lbs/min). Under the terms of the Contract this oil flow rate was selected for all solid-ball and drill-ball bearing tests in Task III.

CONCLUSIONS AND RECOMMENDATIONS

It is concluded that the cage rub experienced by one drilled-ball bearing occurred because of marginal lubrication conditions and is not related to the drilled-ball concept. Therefore, it is recommended that any subsequent evaluation of these solid-ball and drilled-ball bearing designs at high DN levels be conducted at a lubricant supply flow of 61×10^{-3} kilograms per second (8 lbs/min) per bearing or higher to avoid marginal bearing lubrication.

TEST BEARINGS

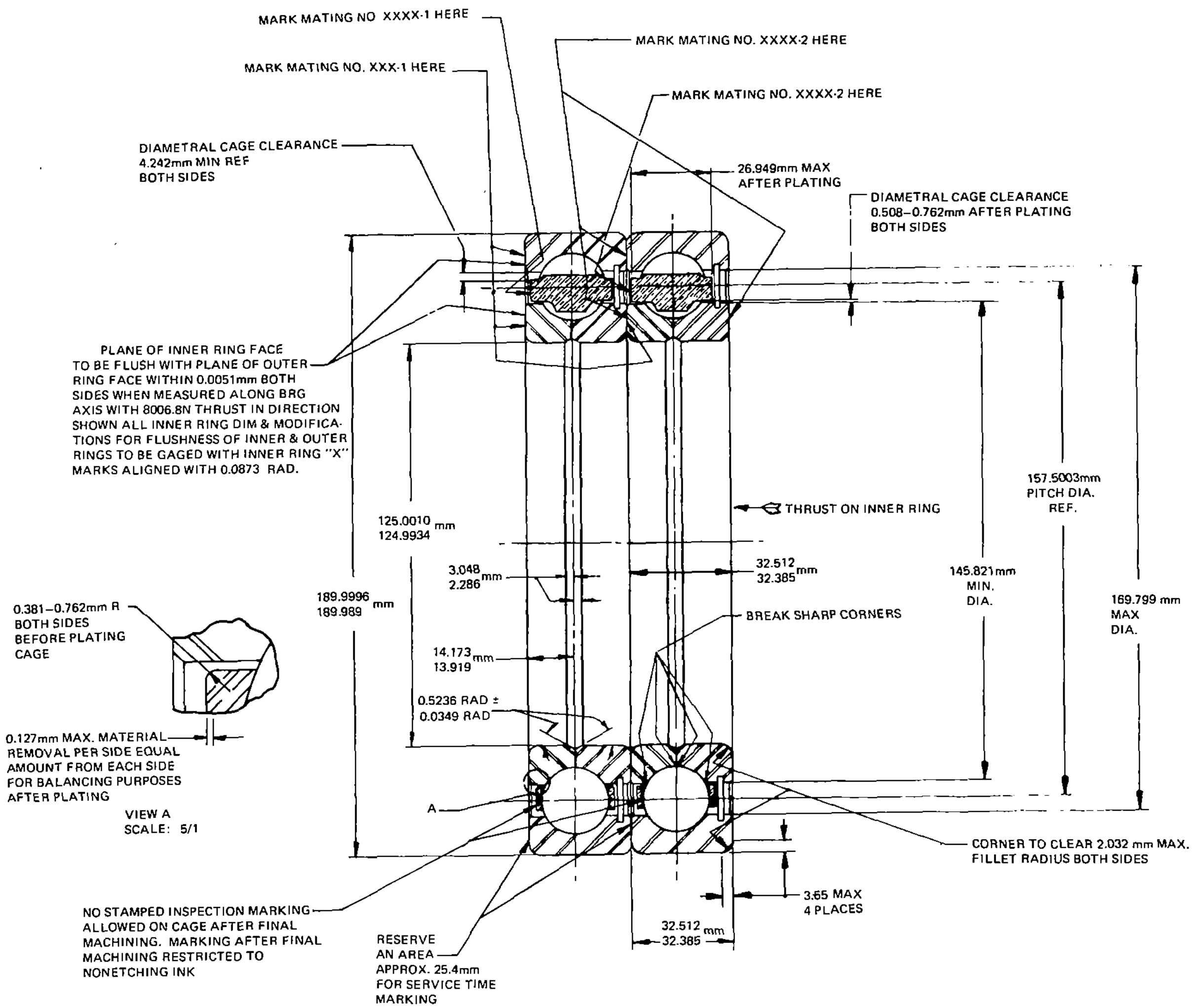
The test bearings utilized in this contract were procured and initially tested at up to 3.0 million DN, under Contract NAS3-13491. The bearings were in good condition after these tests, and required only replating and rebalancing of the cages to be suitable for continued testing under the present contract.

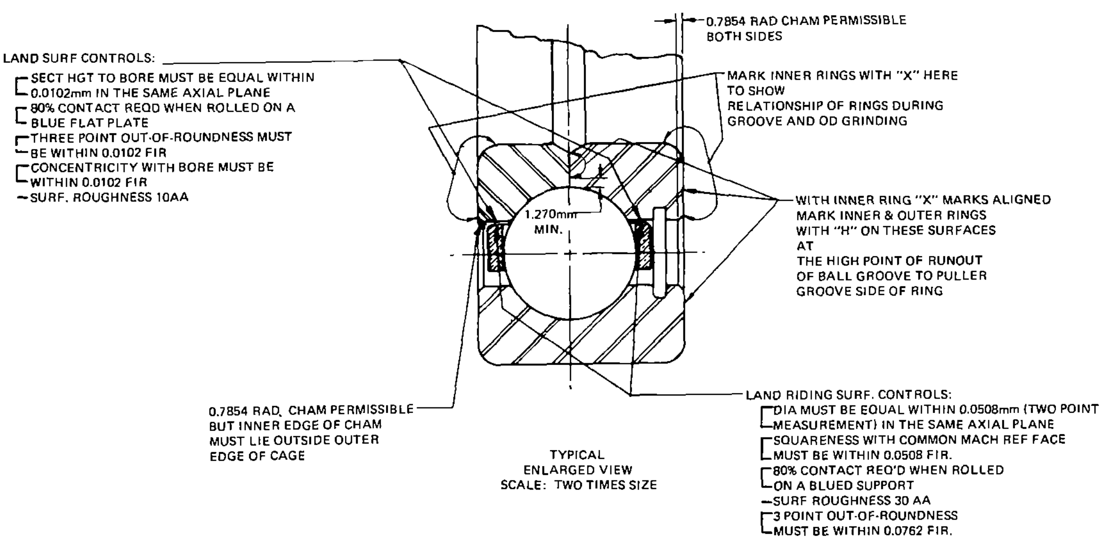
The test bearings were derived from a basic bearing which has been used extensively in a production aircraft engine. The basic bearing is used in pairs to form a duplex thrust bearing in its engine application, as indicated in Figure 1 (1A). It has a bore of 125 mm, a ball diameter of 20.6375 mm (0.8125 in.), a split inner ring, and a one-piece cage which rides on two inner lands. The balls and rings are made of M-50 CVM steel, hardened to Rockwell C60 minimum. The cage is made of AMS 6415 steel, hardened to Rockwell C28-32 and silver plated. The contact angle is 0.4014-0.4538 radians (23-26 degrees) under static conditions with a thrust load of 266.9 newtons (60 pounds). When the bearing is assembled, recesses in the mating surfaces of the two inner rings form lubricant passages between the bearing ID and the inner-race surface. The bearing is made to ABEC 7, and is manufactured by the Marlin-Rockwell Company.

The basic bearing was designed for normal service at about 1.5 million DN, and two modifications were required to ensure satisfactory operation at speeds up to 3.0 million DN. The first modification consisted of installing additional oil passages to the inner rings to improve ring-to-cage lubrication. These passages are shown in Figures 2 and 3. The second modification increased ball pocket diameter to the dimensions shown in Figure 4. This minimizes rubbing between the cage and inner rings due to drag caused by restricted ball travel within the cage pocket.

Under Contract NAS3-13491, eight bearings had been modified in accordance with Figures 2, 3, and 4. Four of these modified bearings were used for baseline testing. The remaining four were altered further to a drilled-ball configuration in which the ball mass was reduced by approximately 50% and stub pins were installed in the cage to prevent the drilled balls from presenting their edges to the bearing races. The ball modifications are shown in Figure 5. Details of the restraining pins are shown in Figure 6, and modifications made in the cage rails to accept the pins are shown in Figure 7. The appearance of the drilled balls and the cages with pins installed is shown in Figures 8 and 9, respectively.

All eight test bearings were in good condition after the testing done under Contract NAS3-13491, and required only replating and rebalancing of the cages to be suitable for continued testing under the present contract. Two of the baseline bearings and two of the drilled-ball bearings were used for the work described in this report.





TYPICAL
ENLARGED VIEW
SCALE: TWO TIMES SIZE

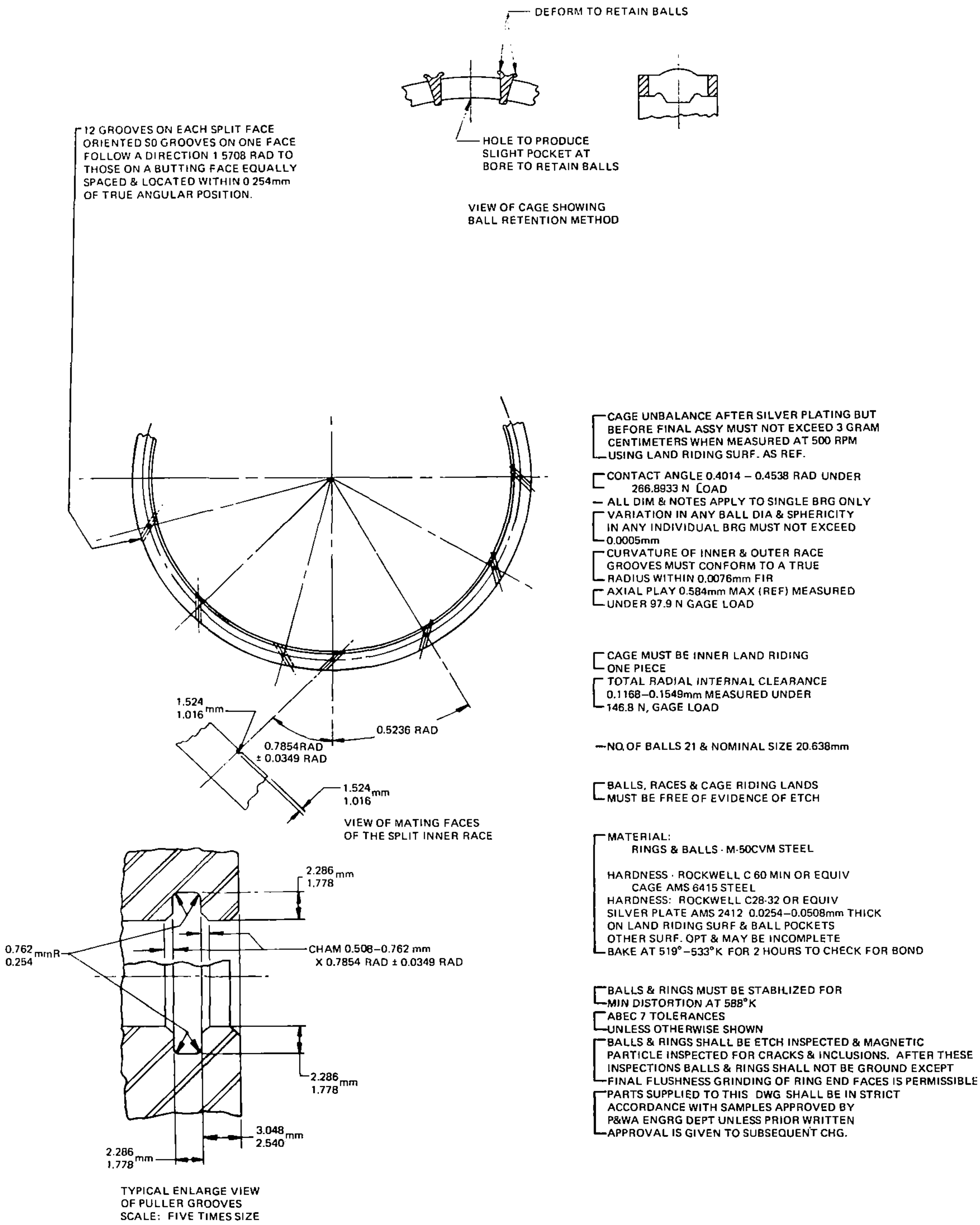
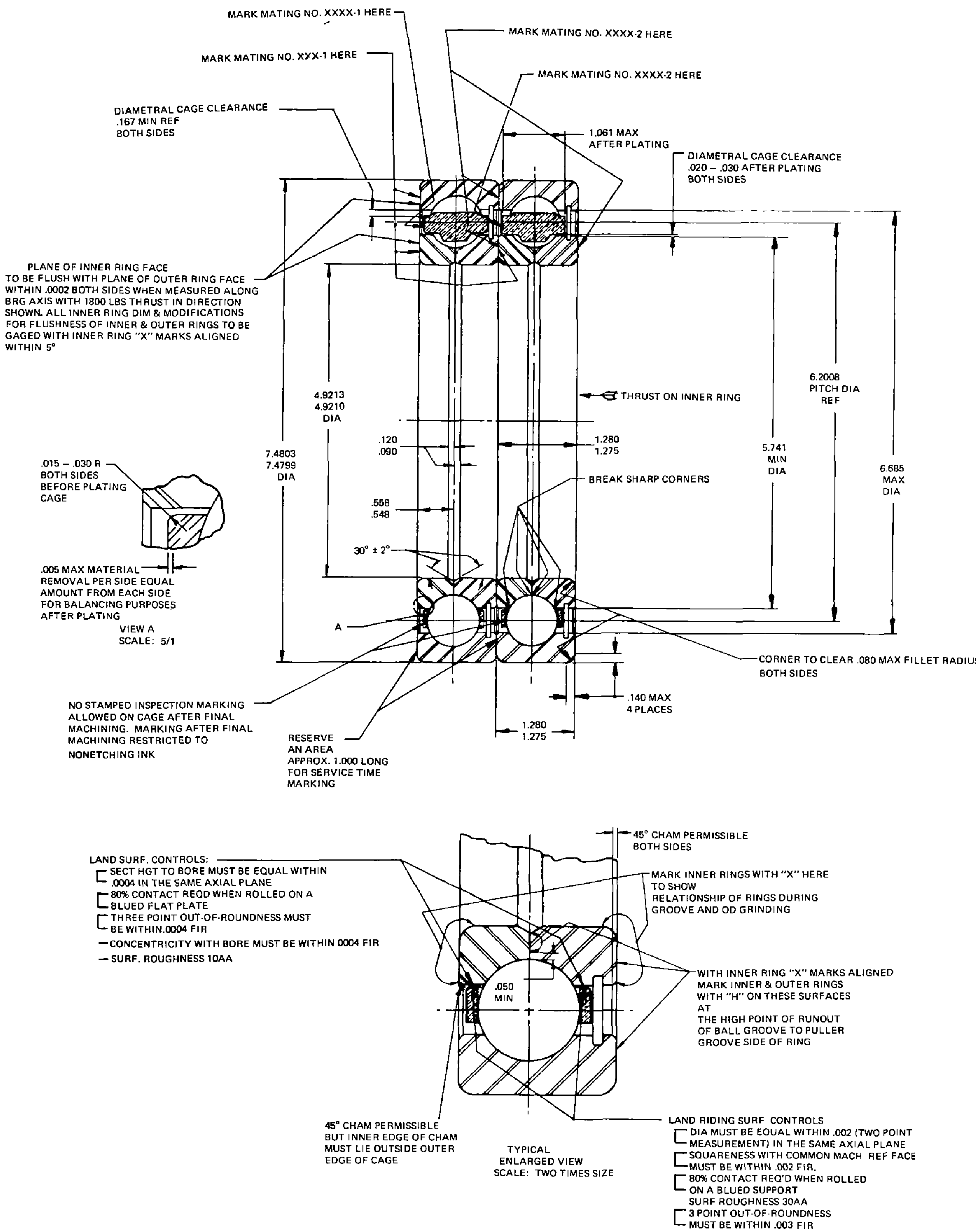


Figure 1
Specification Drawing for Selected Bearing
(Dimensions in millimeters unless otherwise noted)


MARK MATING NO. XXXX-1 HERE
MARK MATING NO. XXXX-2 HERE
MARK MATING NO. XXX-1 HERE
MARK MATING NO. XXXX-2 HERE
DIAMETRAL CAGE CLEARANCE .167 MIN REF BOTH SIDES
1.061 MAX AFTER PLATING
DIAMETRAL CAGE CLEARANCE .020 – .030 AFTER PLATING BOTH SIDES
PLANE OF INNER RING FACE TO BE FLUSH WITH PLANE OF OUTER RING FACE WITHIN .0002 BOTH SIDES WHEN MEASURED ALONG BRG AXIS WITH 1800 LBS THRUST IN DIRECTION SHOWN. ALL INNER RING DIM & MODIFICATIONS FOR FLUSHNESS OF INNER & OUTER RINGS TO BE GAGED WITH INNER RING "X" MARKS ALIGNED WITHIN 5°
6.2008 PITCH DIA REF
4.9213 4.9210 DIA
THRUST ON INNER RING
.120 .090
1.280 1.275
7.4803 7.4799 DIA
5.741 MIN DIA
6.685 MAX DIA
.015 – .030 R BOTH SIDES BEFORE PLATING CAGE
BREAK SHARP CORNERS
.558 .548
30° ± 2°
.005 MAX MATERIAL REMOVAL PER SIDE EQUAL AMOUNT FROM EACH SIDE FOR BALANCING PURPOSES AFTER PLATING
VIEW A SCALE: 5/1
A
CORNER TO CLEAR .080 MAX FILLET RADIUS BOTH SIDES
.140 MAX 4 PLACES
NO STAMPED INSPECTION MARKING ALLOWED ON CAGE AFTER FINAL MACHINING. MARKING AFTER FINAL MACHINING RESTRICTED TO NONETCHING INK
RESERVE AN AREA APPROX. 1.000 LONG FOR SERVICE TIME MARKING
1.280 1.275
45° CHAM PERMISSIBLE BOTH SIDES
LAND SURF. CONTROLS:
SECT HGT TO BORE MUST BE EQUAL WITHIN .0004 IN THE SAME AXIAL PLANE
80% CONTACT REQD WHEN ROLLED ON A BLUED FLAT PLATE
THREE POINT OUT-OF-ROUNDNESS MUST BE WITHIN.0004 FIR
CONCENTRICITY WITH BORE MUST BE WITHIN 0004 FIR
SURF. ROUGHNESS 10AA
MARK INNER RINGS WITH "X" HERE TO SHOW RELATIONSHIP OF RINGS DURING GROOVE AND OD GRINDING
.050 MIN
WITH INNER RING "X" MARKS ALIGNED MARK INNER & OUTER RINGS WITH "H" ON THESE SURFACES AT THE HIGH POINT OF RUNOUT OF BALL GROOVE TO PULLER GROOVE SIDE OF RING
45° CHAM PERMISSIBLE BUT INNER EDGE OF CHAM MUST LIE OUTSIDE OUTER EDGE OF CAGE
TYPICAL ENLARGED VIEW SCALE: TWO TIMES SIZE
LAND RIDING SURF CONTROLS
DIA MUST BE EQUAL WITHIN .002 (TWO POINT MEASUREMENT) IN THE SAME AXIAL PLANE
SQUARENESS WITH COMMON MACH REF FACE MUST BE WITHIN .002 FIR.
80% CONTACT REQ'D WHEN ROLLED ON A BLUED SUPPORT SURF ROUGHNESS 30AA
3 POINT OUT-OF-ROUNDNESS MUST BE WITHIN .003 FIR

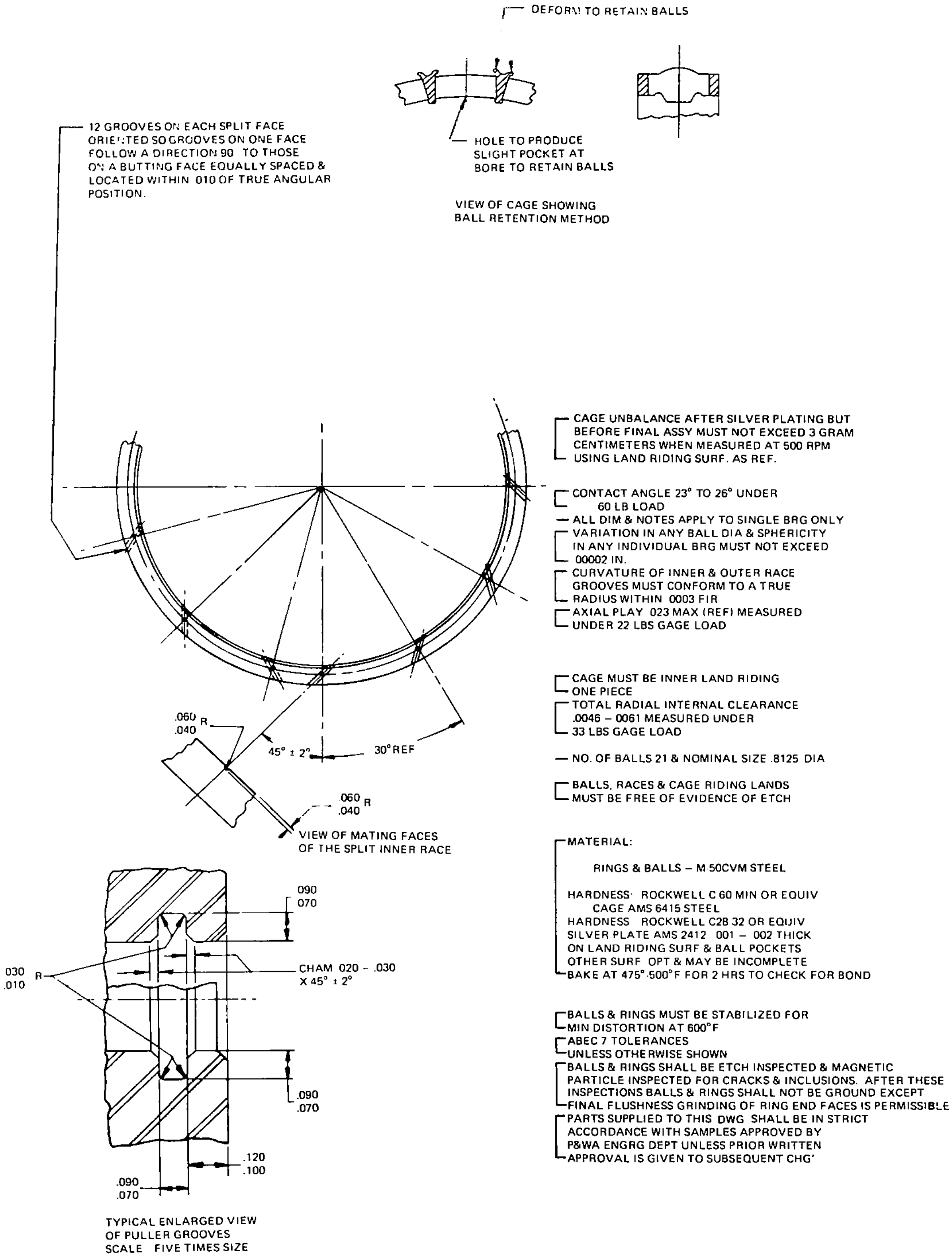


Figure 1A
Specification Drawing for Selected Bearing
(Dimensions in inches unless otherwise noted)

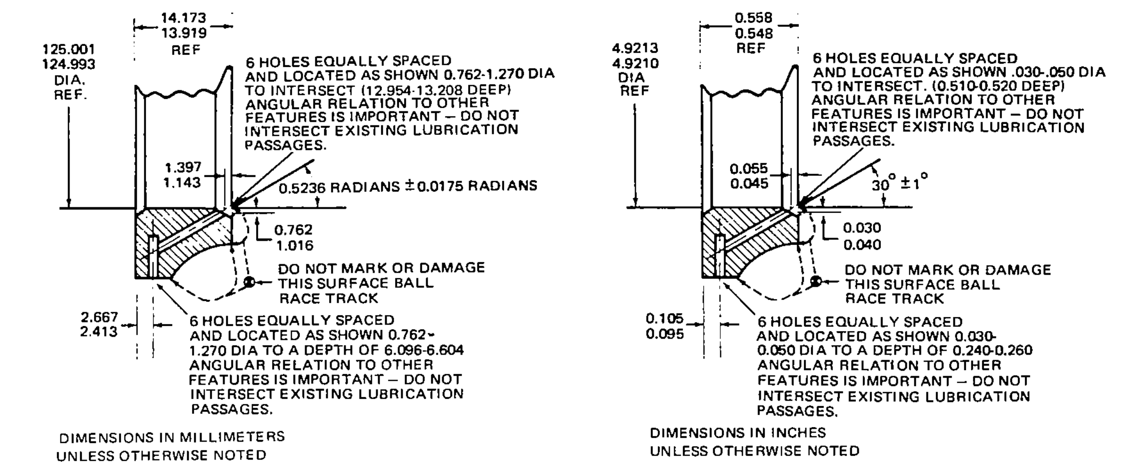


Figure 2 Inner Ring Alteration

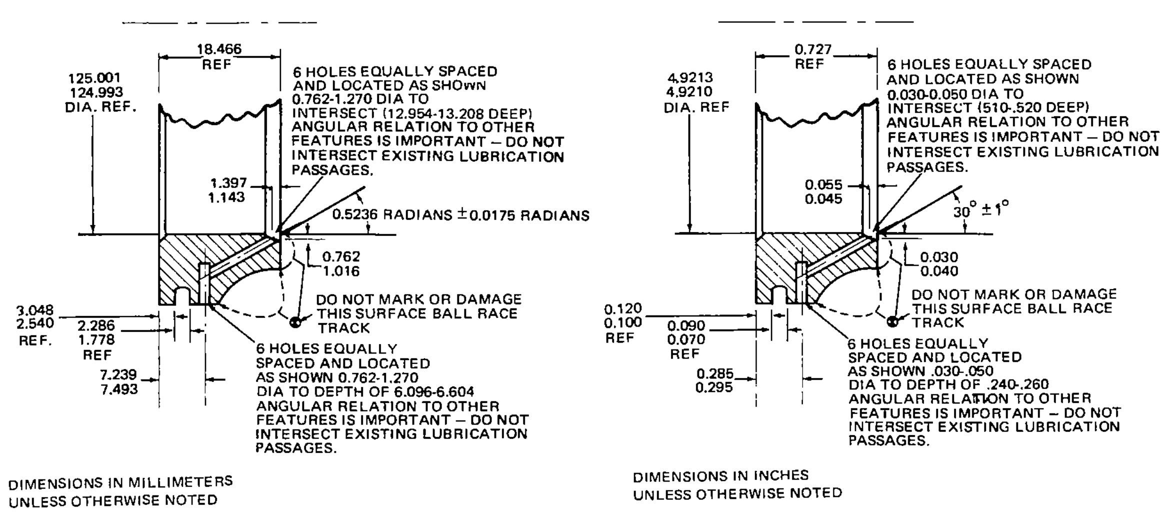
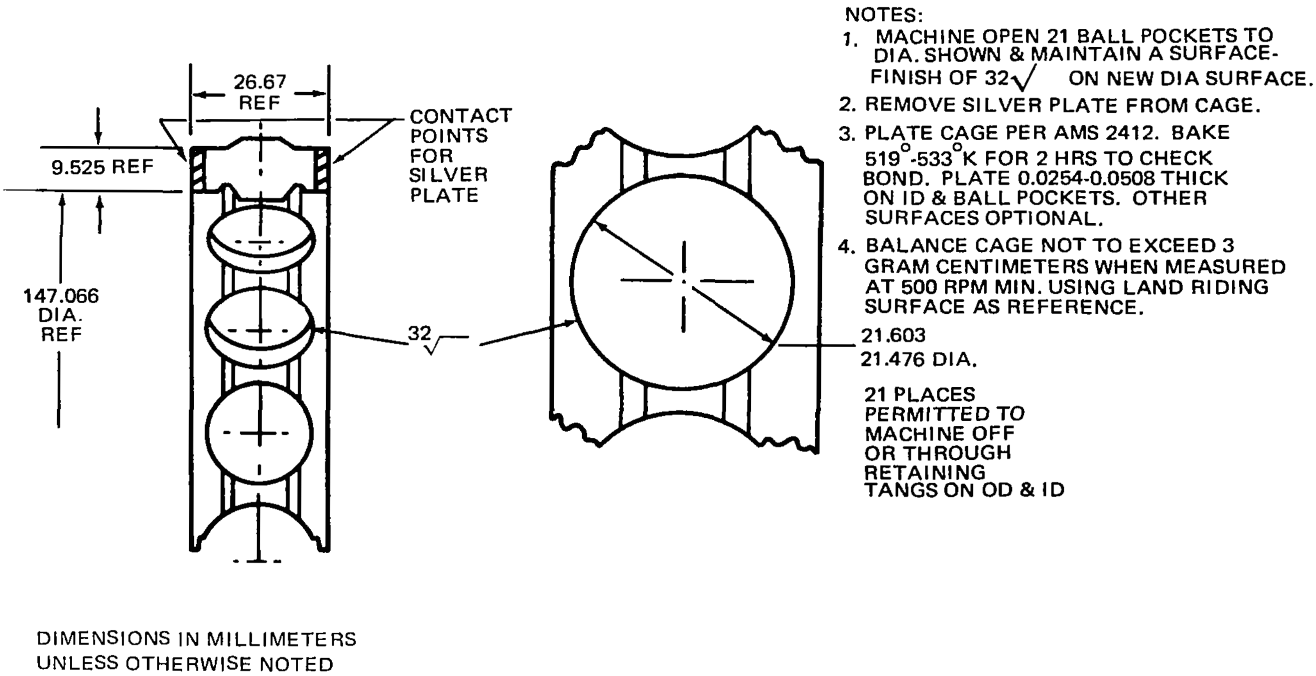


Figure 3 Inner Ring Alteration



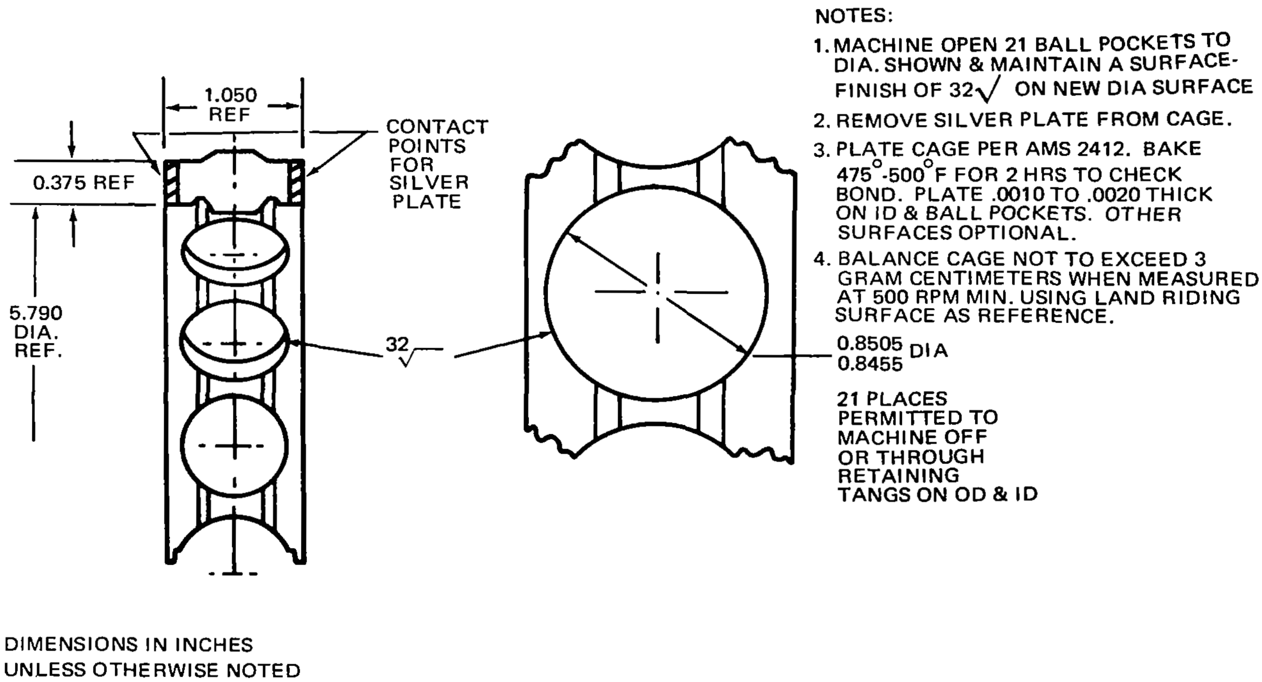
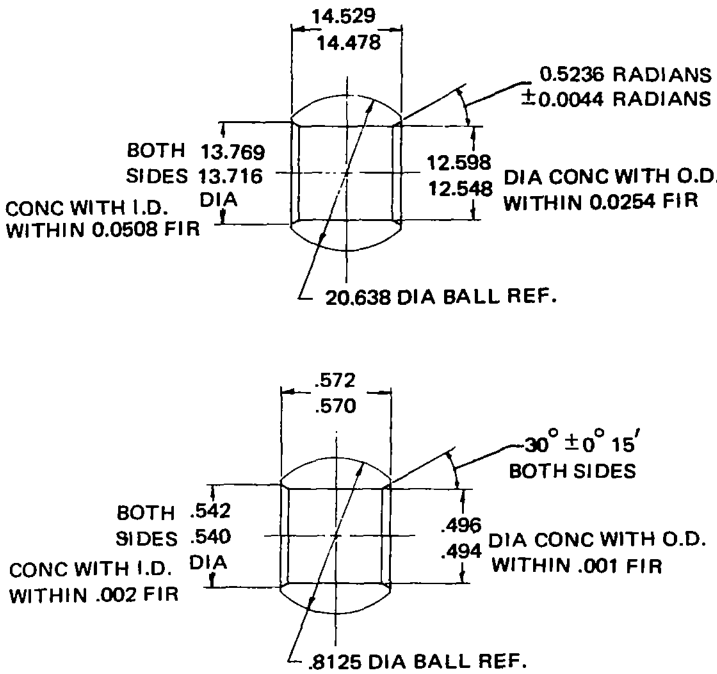


Figure 4 Cage Alteration



DIMENSIONS IN MILLIMETERS
UNLESS OTHERWISE NOTED

DIMENSIONS IN INCHES
UNLESS OTHERWISE NOTED

Figure 5 Ball Detail

DIMENSIONS IN MILLIMETERS
UNLESS OTHERWISE NOTED

DIMENSIONS IN INCHES
UNLESS OTHERWISE NOTED

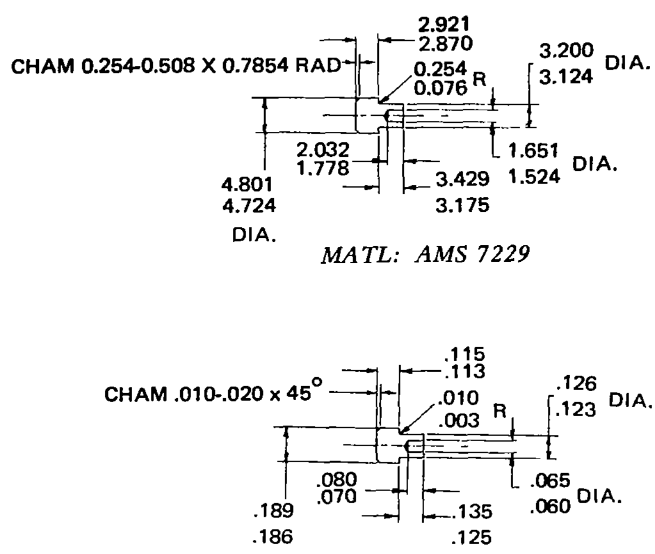


Figure 6 Pin Detail

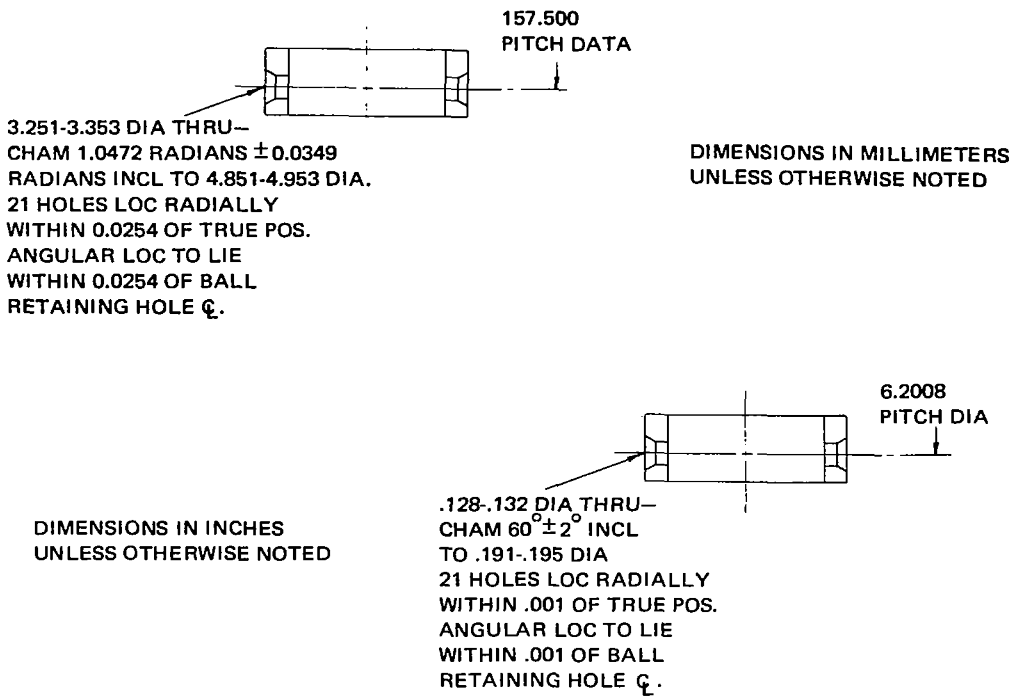


Figure 7 Cage Detail

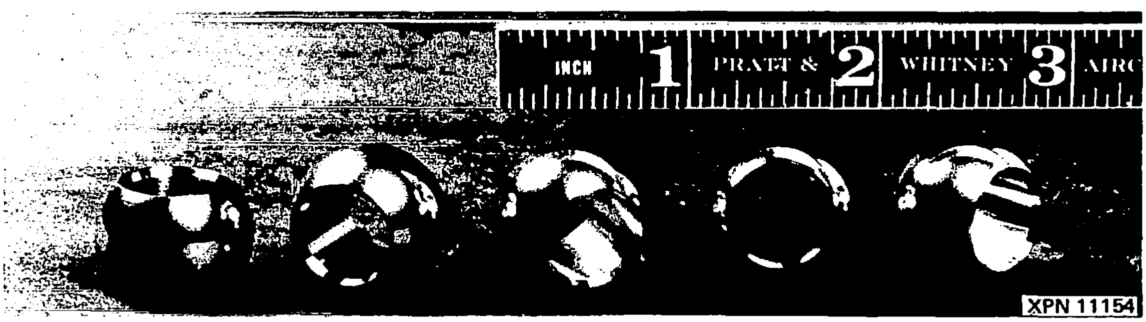


Figure 8 Typical Drilled Balls

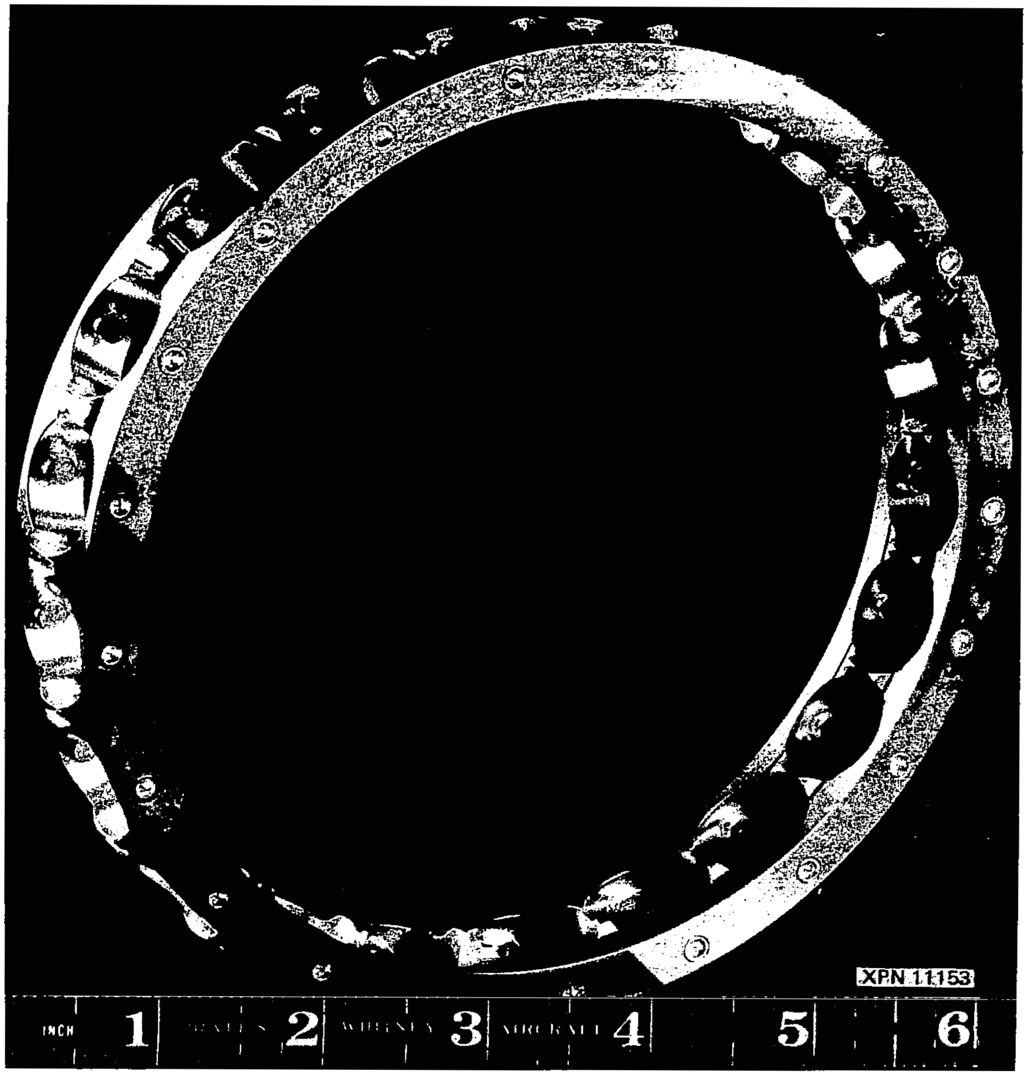


Figure 9 Drilled-Ball Bearing Cage with Pins Installed

TEST EQUIPMENT

Evaluation of the test bearings was carried out in an existing contractor-owned thrust ball bearing rig which had previously demonstrated high DN operational capability. Contractor-owned test stand facilities, which had been used in Contract NAS3-13491, were utilized to produce the operating conditions of the test program. Instrumentation generally used in conventional test practice was employed to measure, monitor, and record test bearing parameters at each operating condition.

BEARING TEST RIG

The bearing test rig, shown in Figure 10, consisted of a cylindrical housing with an annular thrust loading system at one end. Two similar bearings were mounted on a common shaft along with their outer race carriers. This complete assembly was slipped into the cylindrical housing, engaged to a spline on the gearbox drive shaft, and secured with a cover at the front end of the rig. In operation, the hydraulic loading piston applied axial loads to the outer race carrier of the rear bearing. The load was transmitted through the rear bearing to the common shaft, and then through the front bearing to the housing. As a result, two identical bearings were tested simultaneously under identical conditions. The test rig shaft was driven by a variable-speed, 111.9 kilowatt (150 hp) DC electric motor through a 7 to 1 gear speed increaser as shown in Figure 11.

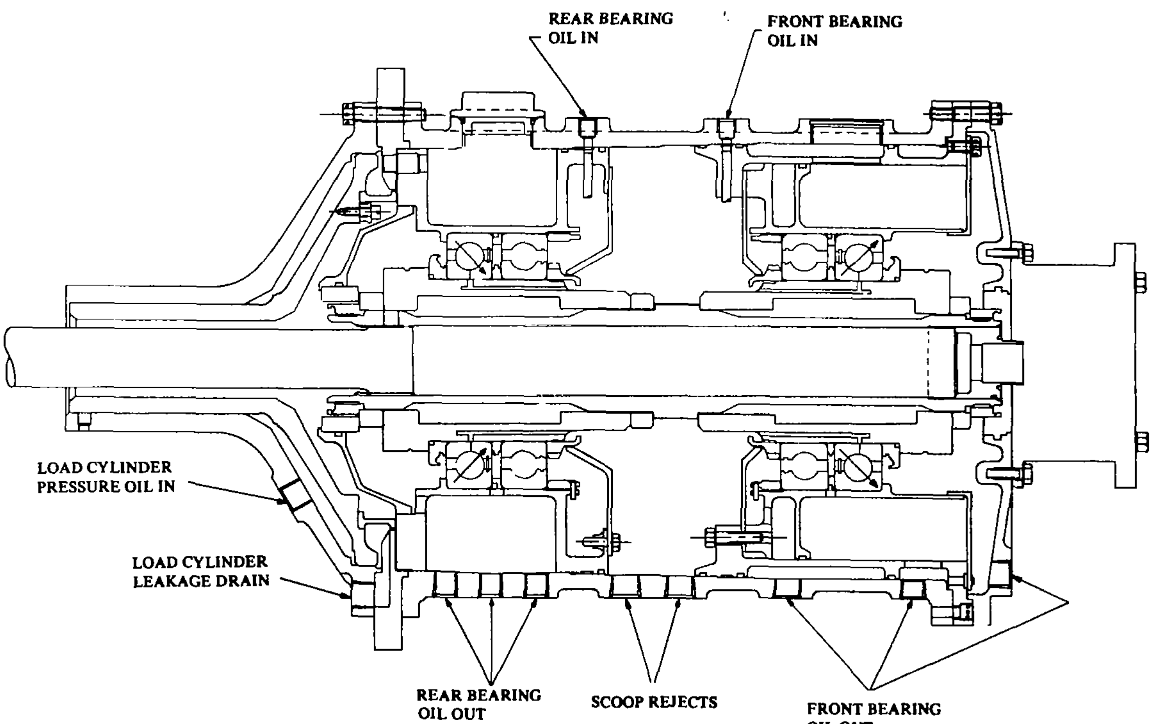


Figure 10 Thrust Bearing Test Rig

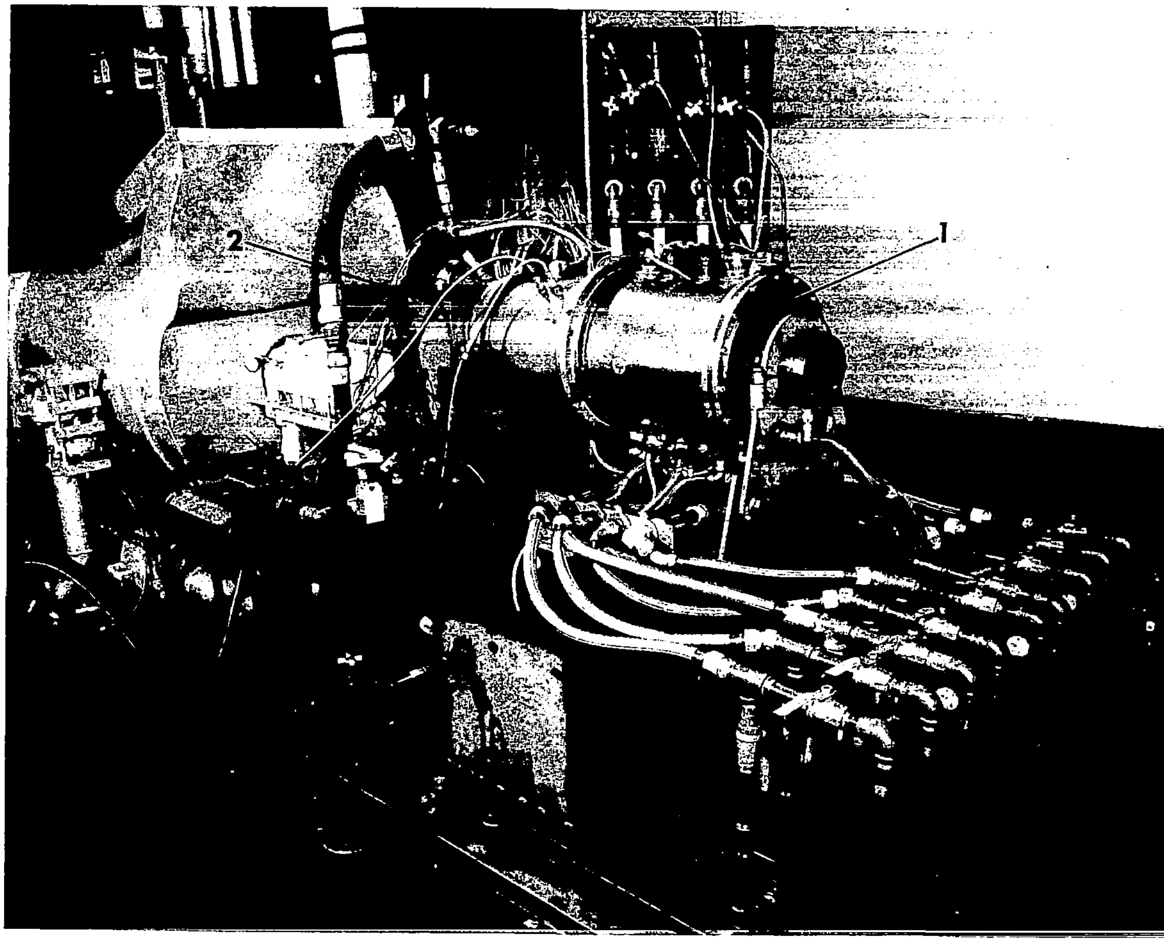
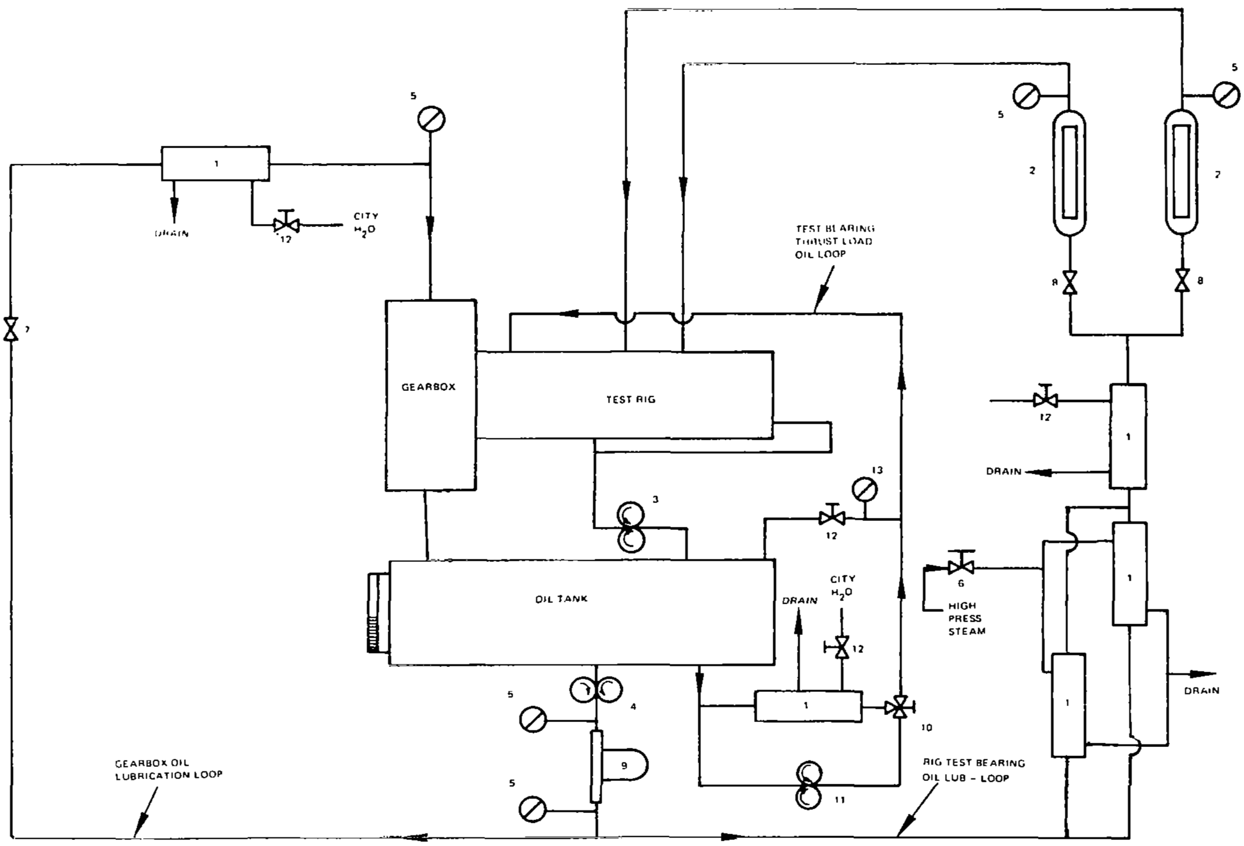


Figure 11 Thrust Bearing Test Rig
(1) Test Rig (2) Gearbox
(3) Electric Drive Motor

LUBRICATION SYSTEM

The relationship between the test rig and the external lubrication system is shown in Figure 12. The complete system consisted of three oil-circulating loops supplied from a common reservoir. One loop lubricated the gearbox for the rig's drive unit. The second loop provided hydraulic pressure for thrust loading the test bearings. The third loop provided lubrication for the test bearings, and contained heat exchangers, instrumentation, and controls necessary for maintaining suitable oil temperatures and flow rates to the test bearings.



1. Heat Exchanger "American Std"
   Series 503, Single Pass

2. Flowmeter "Fisher Porter"
   Model 10A1152AOM, Sign 5

3. Pump "Viking Pump Co"
   No. HL-154, Cap. 11.4 x 10^{-4} m^3/sec (18 gpm)

4. Pump "Viking Pump Co"
   No. 253, Cap. 3.2 x 10^{-4} m^3/sec (5 gpm)

5. Gage "Halicoid"
   Type 440 RH Heliy, Size 4½ 11,
   0-689, 476 N/m^2 (0-100 psi)

6. Control Valve "Jenkins Valve Co"
   No. CF-TM 34-200

7. Pressure Regulating Valve "Cash-Acme"
   Type G-60

8. Valve "March Instrument Co"
   Type 1900 PM-FAA

9. Static Filter "Cuno Co"
   S. S. Metal Mesh, 10 Micron Cap.

10. Pressure Regulating Valve "Cash-Acme"
    Type FR-½ NPT

11. Pump "Viking Pump Co"
    No. FH-54,
    Cap. 1.9 x 10^{-4} m^3/sec (3 gpm)

12. Control Valve "Masoneilan"
    No. DR38-26471

13. Gage "Heise"
    Type H470, 8½ Size
    0-861.845 N/m^2 (0-125 psi)

Figure 12 Lubrication System

TEST BEARING LUBRICATION

The method of supplying lubricant to each bearing is shown in Figure 13. Lubricant from the external reservoir was directed by a fixed nozzle into an annular scoop which rotated with the bearing shaft assembly. The lubricant then flowed from the scoop through axially oriented passages in the hub assembly to radial passages terminating at the bearing bore. Lubricant discharged from the test bearings was collected in manifolds at the bottom of the test rig and returned to the external system.

The test bearings were lubricated through a number of passages which originated at the bearing ID. Twenty-four passages led directly from the bearing bore to the inner-race surface along the parting plane of the split inner ring. Each inner ring had six additional passages leading from the bore to the land surface on which the cage rides. This combination of passages provided lubricant directly to the balls and to the cage riding-surfaces.

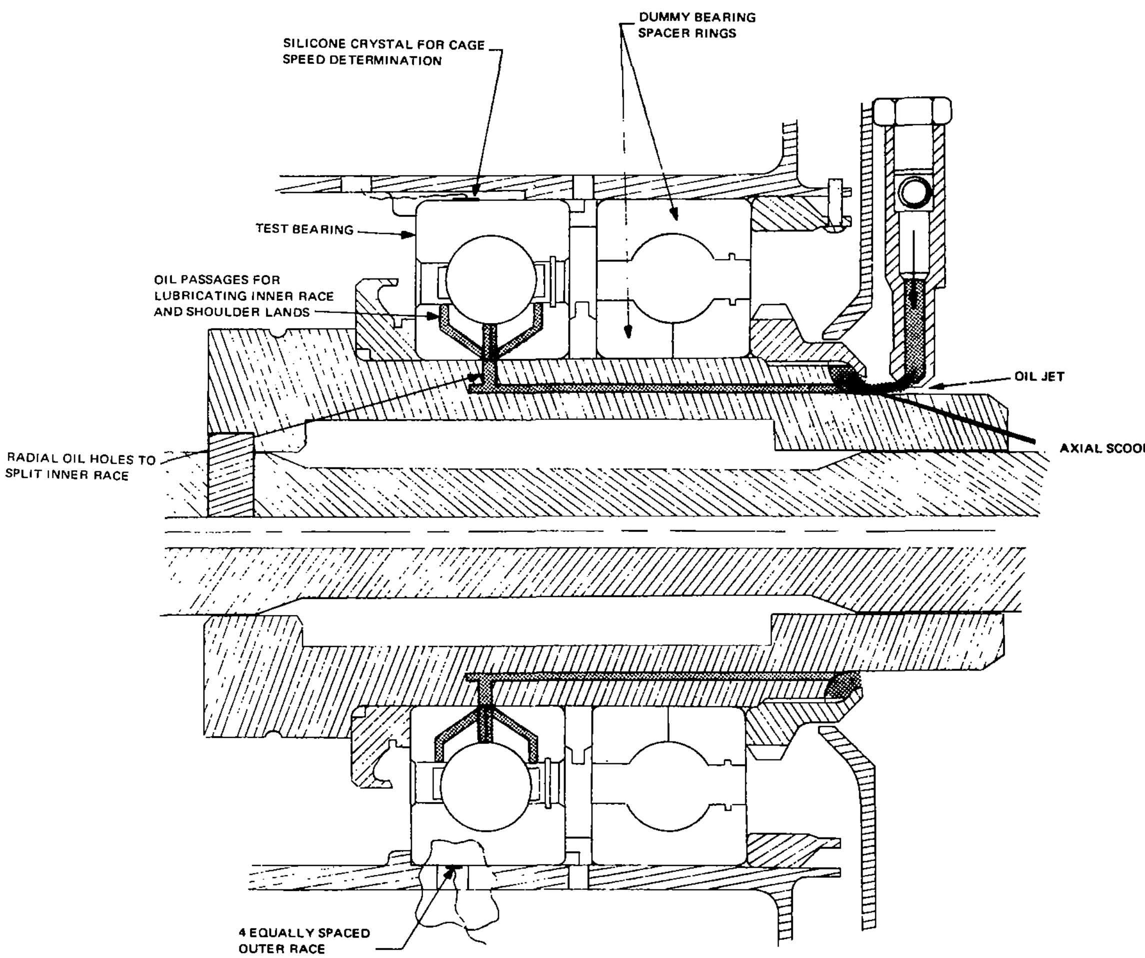


Figure 13 Bearing Lubrication and Instrument Scheme

LUBRICANT

The lubricant used for this program was a polyester oil which conforms to MIL-L-23699-A. The lubricant has been used extensively in rig and engine testing at P&WA, and in the field service operation of P&WA engines. It was used, also, for testing of the eight bearings under Contract NAS3-13491. The characteristics of this lubricant are presented in Table I.

TABLE I
LUBRICANT CHARACTERISTICS

Kinematic Viscosity:		
At 233.2°K; - 40°F	1.3×10^{-2} meter^2/sec (Max.);	13,000 Centistokes (Max.)
310.9°K; 100°F	1.0×10^{-4} meter^2/sec (Max.);	100 Centistokes (Max.)
372.0°K; 210°F	5.5×10^{-6} meter^2/sec (Max.);	5.5 Centistokes (Max.)
477.6°K; 400°F	1.0×10^{-6} meter^2/sec (Max.);	1.0 Centistokes (Max.)
Flash Point:	477.6°K; 400°F (Min)	
Evaporation Loss After 6½ Hours:		
At Sea Level, 477.6°K; 400°F	25% (Max.)	
12,192 meters, 477.6°K; 40,000 ft, 400°F	50% (Max.)	
Gear Scuffing Load:	420.3 newtons/mm; 2400 lb/ in. (Min.)	
Pitting Fatigue	100 hours (Min.)	
Change From Original Viscosity at 310.9°K; 100°F: (After 72 Hours at 448.2°K; 347°F)	-5 to +15%	
Change From Original Total Acid Number: (After 72 Hours at 448.2°K; 347°F)	2.0 (Max.)	
Quality:	Lubricant free of suspended matter, grit water, and objectionable odor.	

TEST MEASUREMENTS

The parameters measured during the test program and the accuracies attained are listed in Table II.

TABLE II
TEST MEASUREMENTS

Thrust Load	**0.5% of full-scale gage reading**
Shaft Speed	**0.5% of measured speed**
Oil-Flow Rate	**1.0% of full-scale meter reading**
Oil-in Temperature	**±1.1°K up to 549.8°K; ±2°F up to 530°F**
Oil-out Temperature	**±1.1°K up to 549.8°K; ±2°F up to 530°F**
Outer Race Temperature	**±1.1°K up to 549.8°K; ±2°F up to 530°F**
Bearing Cage Speed	**±0.2% of measured speed**
Rig Vibration	**2.0% of full-scale meter reading**

All pertinent rig and bearing temperatures were recorded on a multichannel, 255.4 - 588.7°K (0-600°F), Alumel-Chromel, Bristol Flight Recorder which provided a permanent record with a complete set of temperature data taken every 15 seconds.

Alumel-Chromel thermocouples were immersed in the lubrication system to monitor rig oil-in and oil-out temperatures. Four equally-spaced Alumel-Chromel thermocouples were tack-welded to each outer race OD surface to measure bearing temperatures as illustrated in Figure 13.

THERMAL STABILITY CRITERIA

A supplementary thermocouple circuit, shown in Figure 14, was utilized to determine bearing thermal stability at constant operating conditions after a test point was set. Thermal stability was assumed to exist when the difference between outer-race temperature and oil-in temperature did not change by more than 1.1°K (2°F) in a period of five minutes. This approach was used because changes in oil-in temperature are reflected quickly in corresponding changes in outer-race temperature. Both temperatures can change slightly over a period of minutes as a result of practical limitations of the oil temperature control system, even though bearing heat generation has stabilized. Stabilization of the differential temperature measurement provided a direct indication that bearing heat generation had stabilized.

Since the test rig imposed substantially identical operating conditions on both test bearings simultaneously, the supplementary circuit was attached only to the front bearing during each test sequence. Variation in the differential temperature was easily interpreted to within 0.06°K (0.10°F) on a Bristol 760 Recorder. A typical stabilization chart for a five minute period is shown in Figure 15. The absolute difference between outer-race and oil-in temperature for the bearing in the rear rig position was recorded in millivolts on the same Bristol 760 Recorder.

BEARING CAGE SPEED

A special technique developed at Pratt & Whitney Aircraft was used to measure the bearing cage speed without affecting bearing operation. The measurement was made with a micro-measurement DGP-1000-500 semi-conductor strain gage attached to the bearing outer-race (Figure 13). The gage senses the change in outer-race strain as each ball passes the gage site, and produces signals at a rate proportional to cage speed and the number of balls in the bearing. The measurements are very accurate and provided a means for detecting ball skidding.

RIG VIBRATION

Rig vibration was monitored by bearing failure indicators developed by Pratt & Whitney Aircraft to detect abnormal bearing operating conditions. With this sensitive instrumentation, an increase of vibration would have indicated ball or race spalling at its inception, making it possible to terminate the test before gross damage occurred. This is an important aid in any failure analysis, and minimizes the possibility of rig damage and wasted test effort.

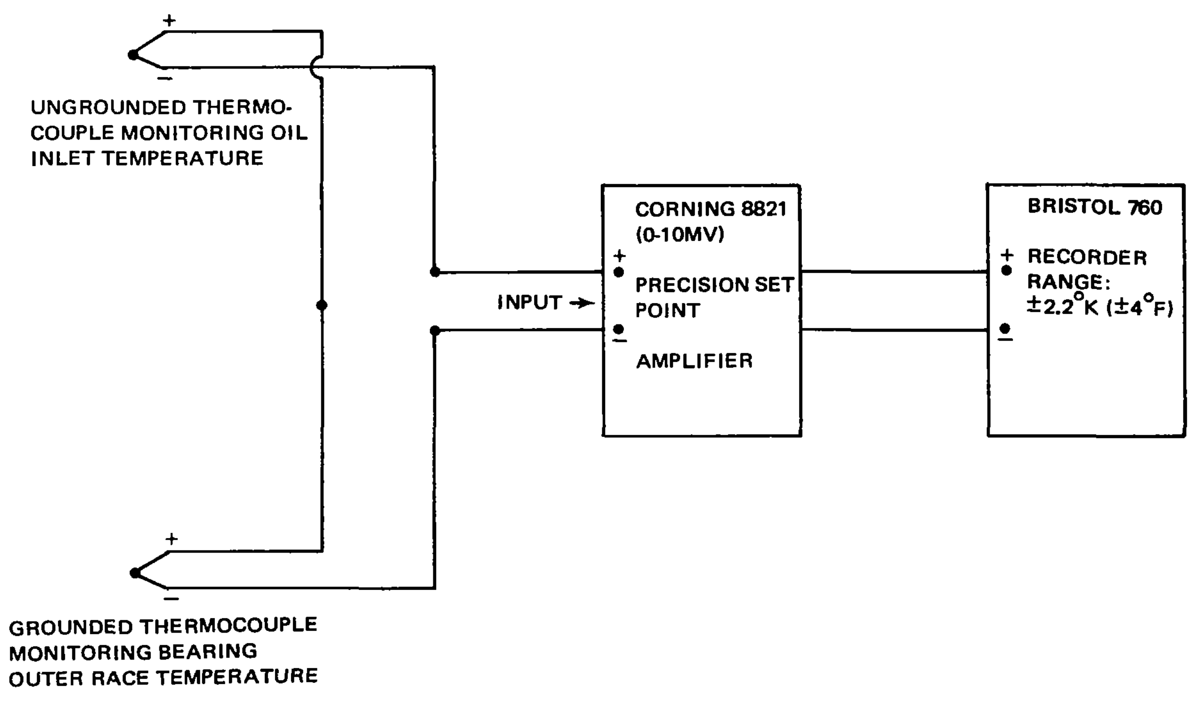


Figure 14 Supplementary Thermocouple Circuit

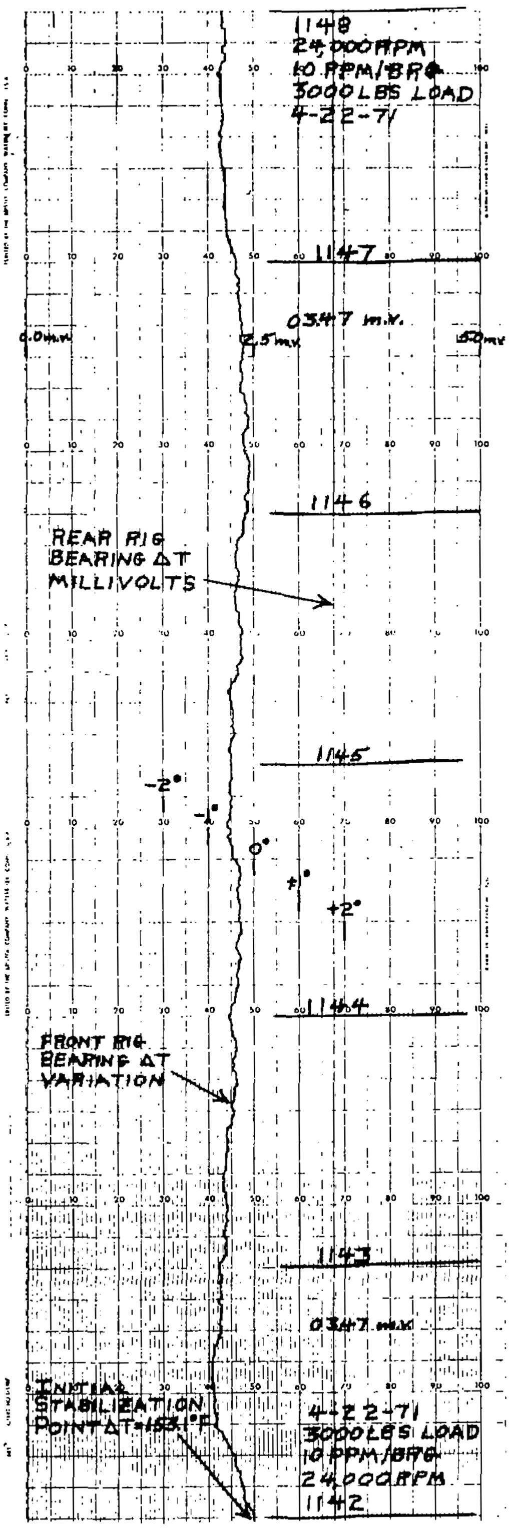


Figure 15 Typical ΔT Stabilization Chart

TASK I OIL SCOOP CALIBRATION STUDY

Lubrication of the test bearings is provided through a combination of passages originating at the bearing ID. These passages supply lubricant directly to the balls and to the cage riding-surfaces of the inner rings. The method of supplying lubricant to the bore of each bearing is shown in Figure 13. Lubricant is directed by a fixed jet into an annular scoop which rotates with the bearing shaft assembly. The lubricant accepted by the scoop flows from the scoop through axially oriented passages in the hub assembly to radial passages terminating at the bearing bore. With this technique of transporting lubricant to the test bearing, the oil flow rate into the bearing depends upon the lubricant flow supplied by the fixed jet, the effectiveness of the scoop in capturing the jetted oil, and the fluid pumping capability of the hub/bearing passages. Since the oil supplied by the jet can be controlled, it is important to determine the efficiency of the scoop-hub assembly in collecting and transporting the oil to the bearing. This was done under Task I with contractor-owned solid-ball bearings as described below.

TEST CONDITIONS

The selected test operating speeds and oil supply rates were representative of those to be used subsequently in the performance evaluation of the solid-ball and drilled-ball bearings. They are summarized in Table III.

TABLE III
TEST RIG SCOOP CALIBRATION OPERATING CONDITIONS

Thrust load: 11,121 newtons per bearing; 2,500 lbs per bearing

Temperature: Oil-in 366.5°K ± 1.7°K; 200°F ± 3°F

OIL-JET FLOW RATE		APPROXIMATE BEARING OIL SUPPLY*	
(kg/sec/jet)	(lbs/min/jet)	(kg/sec/bearing)	(lbs/min/bearing)
161×10^{-3}	21.3	121×10^{-3}	16
141	18.7	106	14
121	16.0	90.7	12
101	13.3	75.6	10
61	8.0	45.4	6

*Based on an assumed 75% scoop efficiency

Shaft Speed:		
	8,000 rpm	$(1.0 \times 10^6$ DN)
	12,000	$(1.5 \times 10^6$ DN)
	16,000	$(2.0 \times 10^6$ DN)
	19,200	$(2.4 \times 10^6$ DN)
	20,800	$(2.6 \times 10^6$ DN)
	24,000	$(3.0 \times 10^6$ DN)

The thrust load of 11,121 newtons (2500 lbs) per bearing was maintained throughout the entire scoop-calibration program. Oil was supplied at a nominal temperature of 366.5°K (200°F) since this oil supply temperature was to be used in much of the subsequent evaluation of baseline and drilled-ball bearing performances. Previous testing at Pratt & Whitney Aircraft had shown that the scoop efficiencies of the rig were normally in the range of 70% to 80%; that is, 70% to 80% of the jetted oil was captured and transported to the test bearings. A scoop efficiency of 75% was assumed in establishing the oil-jet flow rates of Table III so that the desired values of bearing oil supply could be approximated.

TEST PROCEDURE

Calibration testing was initiated at 161 x 10^{-3} kilograms per second (21.3 lbs/min) per jet and then progressed to successively lower oil-jet flow rates. At each flow rate, rig shaft speed was set initially at 8,000 rpm and subsequently increased to 24,000 rpm as shown in Table III. Since scoop efficiency is the percentage of jetted oil captured and transported into the bearing, two or three consecutive sets of oil weights were taken of all oil discharged from the rig at each test point to ensure an accurate determination of scoop-rejected lubricant and lubricant passed through the bearing.

TEST RESULTS

Calibration of the two oil-scoops revealed that their performances were identical throughout the range of speeds and flows investigated. It was determined that a nominal efficiency of 76% ± 3% existed over the full range of oil-jet supply flows at shaft speeds of 12,000 rpm and above. At 8,000 rpm the nominal efficiency was found to be 76% ± 3% up to a jet supply of 121 x 10^{-3} kilograms per second (16 lbs/min). Above this flow rate, scoop efficiency decreased as shown in Tables IV and V to 68% at 141 x 10^{-3} kilograms per second (18.7 lbs/min) and to 58% at 161 x 10^{-3} kilograms per second (21.3 lbs/min).

The decrease in scoop performance at low speed and high oil-jet flow rates reflects the fact that the lubricant passages in the hub and bearing effectively pump lubricant from the scoop cavity. The amount of lubricant that can be pumped, and which the scoop can accept from the oil-jet, is largely determined by shaft speed. As long as the hub/bearing pumping capacity is large enough, the scoop accepts some fraction (nominally 76%) of all the oil directed at it. When the pumping capacity is small, as at low speeds, and the oil-jet flow is large, the scoop runs full and simply rejects the oil that the hub and bearing passages cannot pump; this results in an apparent reduction in scoop efficiency at low speeds and large oil-jet flow rates.

Bearing temperature and cage speed data obtained during the calibration test series are presented in Tables IV and V. The oil-inlet temperature was held at 366.5°K ±1.7°K (200°F ±3°F) over the range of oil-jet supply flows and shaft speeds. At each oil-jet supply flow, the outer-race average temperatures of both bearings (S/N 2596 A-1 and S/N 2596 A-2) increased almost linearly with an increase of shaft speed. The rear bearing ran somewhat warmer–from 1.1°K (2°F), at all oil flows and 8,000 rpm, to 21.7°K (39°F) at the lowest jet flow of 61 x 10^{-3} kilograms per second (8.0 lbs/min) and 24,000 rpm.

TABLE IV

OIL SCOOP CALIBRATION STUDY

					NO. 2 BEARING (REAR) P/N SKN 52575 S/N 2596A-2					NO. 1 BEARING (FRONT) P/N SKN 52575 S/N 2596A-1				
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)
Thrust Load (newtons)	DN x 10^6	Shaft Speed (rpm)	Jet Oil Flow Rate (10^{-3} kg/sec)	Bearing Oil Flow Rate (10^{-3} kg/sec)	Cage Speed Percent *	Oil Inlet Temp (°K)	Ave. Outer Race Temp (°K)	Outer Race Oil Inlet ΔT (°K)	Avg. Oil Outlet Temp (°K)	Cage Speed Percent*	Oil Inlet Temp (°K)	Avg. Outer Race Temp (°K)	Outer Race Oil Inlet ΔT (°K)	Running Time (hrs)
11121	1.0	8,000	161	93.4	44.78	366.48	381.48	15.00	377.04	44.74	366.48	381.48	15.00	1.00
11121	1.5	12,000	161	122.5	45.55	366.48	394.26	27.78	388.71	45.55	366.48	393.15	26.67	1.00
11121	2.0	16,000	161	122.5	45.56	366.48	411.48	45.00	404.26	45.26	366.48	408.15	41.67	1.00
11121	2.4	19,200	161	122.5	45.66	366.48	422.04	55.56	414.82	45.33	366.48	419.26	52.78	1.00
11121	2.6	20,800	161	122.5	45.61	366.48	429.26	62.78	423.15	45.90	366.48	427.04	60.56	1.00
11121	3.0	24,000	161	122.5	45.27	366.48	441.48	75.00	436.48	45.58	366.48	438.71	72.23	1.00
11121	1.0	8,000	141	95.8	–	366.48	382.59	16.11	379.82	–	366.48	381.48	15.00	.75
11121	2.0	16,000	141	107.1	45.71	366.48	409.26	42.78	401.48	45.35	366.48	402.59	36.11	.75
11121	2.4	19,200	141	107.1	44.87	366.48	422.04	55.56	415.37	44.34	366.48	419.26	52.78	.75
11121	2.6	20,800	141	107.1	45.41	366.48	431.48	65.00	426.48	44.79	366.48	425.93	59.45	.75
11121	3.0	24,000	141	107.1	45.10	366.48	444.82	78.34	441.48	44.55	366.48	438.71	72.23	.50
11121	1.0	8,000	121	91.9	45.35	366.48	387.04	20.56	382.59	45.39	366.48	385.93	19.45	.75
11121	1.5	12,000	121	91.9	45.31	366.48	402.04	35.56	395.37	45.24	366.48	399.26	32.78	.75
11121	2.0	16,000	121	91.9	45.88	366.48	418.15	51.67	408.71	45.60	366.48	415.93	49.45	.75
11121	2.4	19,200	121	91.9	46.09	366.48	432.59	66.11	423.71	45.50	366.48	428.71	62.23	.75
11121	2.6	20,800	121	91.9	46.08	366.48	437.59	71.11	430.37	45.42	366.48	434.82	68.34	.75
11121	3.0	24,000	121	91.9	45.74	366.48	451.48	85.00	445.37	45.01	366.48	448.15	81.67	.75
11121	2.0	16,000	101	76.5	45.46	366.48	419.82	53.34	412.59	45.03	366.48	416.48	50.00	.75
11121	2.4	19,200	101	76.5	45.97	366.48	435.93	69.45	428.15	45.33	366.48	429.82	63.34	.75
11121	2.6	20,800	101	76.5	45.79	366.48	443.15	76.67	437.04	45.04	366.48	438.15	71.67	.75
11121	3.0	24,000	101	76.5	–	366.48	449.82	83.34	443.15	–	366.48	448.15	81.67	.50
11121	1.0	8,000	61	46.0	45.24	366.48	390.93	24.45	383.71	45.24	366.48	390.37	23.89	1.00
11121	1.5	12,000	61	46.0	45.60	366.48	411.48	45.00	400.37	45.58	366.48	407.59	41.11	1.00
11121	2.0	16,000	61	46.0	40.58	366.48	433.15	66.67	419.82	40.54	366.48	425.93	59.45	1.00
11121	2.4	19,200	61	46.0	45.14	366.48	451.48	85.00	440.37	45.15	366.48	443.15	76.67	1.00
11121	2.6	20,800	61	46.0	43.85	366.48	455.93	89.45	448.15	44.16	366.48	447.04	80.56	1.00
11121	3.0	24,000	61	46.0	43.56	366.48	484.82	118.34	464.82	44.09	366.48	463.15	96.67	1.00

* Cage Speed is Expressed as a Percent of Shaft RPM

TABLE V
OIL SCOOP CALIBRATION STUDY

					NO. 2 BEARING (REAR) P/N SKN 52575 S/N 2596A-2					NO. 1 BEARING (FRONT) P/N SKN 52575 S/N 2596A-1				
(1) Thrust Load (lbs)	(2) DN x 10^6	(3) Shaft Speed (rpm)	(4) Jet Oil Flow Rate (ppm)	(5) Bearing Oil Flow Rate (ppm)	(6) Cage Speed Percent*	(7) Oil Inlet Temp (°F)	(8) Avg Outer Race Temp (°F)	(9) Outer Race Oil Inlet ΔT (°F)	(10) Avg. Oil Outlet Temp (°F)	(11) Cage Speed Percent*	(12) Oil Inlet Temp. (°F)	(13) Avg. Outer Race Temp (°F)	(14) Outer Race Oil Inlet ΔT (°F)	(15) Running Time (hrs)
2500	1.0	8,000	21.3	12.36	44.78	200	227	27	219	44.74	200	227	27	1.00
2500	1.5	12,000	21.3	16.20	45.55	200	250	50	240	45.55	200	248	48	1.00
2500	2.0	16,000	21.3	16.20	45.56	200	281	81	268	45.26	200	275	75	1.00
2500	2.4	19,200	21.3	16.20	45.66	200	300	100	287	45.33	200	295	95	1.00
2500	2.6	20,800	21.3	16.20	45.61	200	313	113	302	45.90	200	309	109	1.00
2500	3.0	24,000	21.3	16.20	45.27	200	335	135	326	44.58	200	330	130	1.00
2500	1.0	8,000	18.7	12.67	–	200	229	29	224	–	200	227	27	.75
2500	2.0	16,000	18.7	14.17	45.71	200	277	77	263	45.35	200	265	65	.75
2500	2.4	19,200	18.7	14.17	44.87	200	300	100	288	44.34	200	295	95	.75
2500	2.6	20,800	18.7	14.17	45.41	200	317	117	308	44.79	200	307	107	.75
2500	3.0	24,000	18.7	14.17	45.10	200	341	141	335	44.55	200	330	130	.50
2500	1.0	8,000	16.0	12.16	45.35	200	237	37	229	45.39	200	235	35	.75
2500	1.5	12,000	16.0	12.16	45.31	200	264	64	252	45.24	200	259	59	.75
2500	2.0	16,000	16.0	12.16	45.88	200	293	93	276	45.60	200	289	89	.75
2500	2.4	19,200	16.0	12.16	46.09	200	319	119	303	45.50	200	312	112	.75
2500	2.6	20,800	16.0	12.16	46.08	200	328	128	315	45.42	200	323	123	.75
2500	3.0	24,000	16.0	12.16	45.74	200	353	153	342	45.01	200	347	147	.75
2500	2.0	16,000	13.3	10.12	45.46	200	296	96	283	45.03	200	290	90	.75
2500	2.4	19,200	13.3	10.12	45.97	200	325	125	311	45.33	200	314	114	.75
2500	2.6	20,800	13.3	10.12	45.79	200	338	138	327	45.04	200	329	129	.75
2500	3.0	24,000	13.3	10.12	–	200	350	150	338	–	200	347	147	.50
2500	1.0	8,000	8.0	6.08	45.24	200	244	44	231	45.24	200	243	43	1.00
2500	1.5	12,000	8.0	6.08	45.60	200	281	81	261	45.58	200	274	74	1.00
2500	2.0	16,000	8.0	6.08	40.58	200	320	120	296	40.54	200	307	107	1.00
2500	2.4	19,200	8.0	6.08	45.14	200	353	153	333	45.15	200	338	138	1.00
2500	2.6	20,800	8.0	6.08	43.85	200	361	161	347	44.16	200	345	145	1.00
2500	3.0	24,000	8.0	6.08	43.56	200	413	213	377	44.09	200	374	174	1.00

*Cage Speed is Expressed as a Percent of Shaft RPM.

A reduction in oil flow produced corresponding increases in outer-race and oil-outlet average temperatures. Oil-outlet temperatures were not obtained for the front bearing (S/N 2596 A-1) as both thermocouples were found to be erratic. Bearing cage speeds ranged between 44.5% and 46.1% of shaft speed for most combinations of shaft speed and oil flows and did not reflect any definite trend. However, reductions in cage speed of several percent were observed at the lowest oil-flow rate, particularly at the higher shaft speeds. It is believed that these lower values of cage speed are related to incipient skidding within the used solid-ball bearings. These contractor-owned bearings had accumulated a considerable amount of running time in previous testing at Pratt & Whitney Aircraft.

TASK II BEARING OIL FLOW RATE STUDY

In Contract NAS3-13491, the performances of solid-ball (baseline) and drilled-ball bearings were evaluated at an oil supply rate of approximately 60.5 x 10^{-3} kilograms per second (8 lbs/min) per bearing up to three million DN. At the completion of the contract, it was apparent that additional performance data were desirable over a range of oil flows and speeds. Therefore, Task II of this contract was undertaken to determine the behavior of solid-ball and drilled-ball bearings at oil supply flows from 121 x 10^{-3} kilograms per second (16 lbs/min) to 45.4 x 10^{-3} kilograms per second (6 lbs/min) up to 3.0 million DN. The results of Task II were to be the basis for selecting a bearing oil-flow rate for subsequent skid-mapping tests (Task III) and cyclic endurance tests (Task IV). The test results from Task III and Task IV will be presented in the Second Topical Report.

TEST CONDITIONS

Two solid-ball bearings and two drilled-ball bearings were evaluated at the operating conditions summarized in Table VI.

TABLE VI
OIL FLOW RATE STUDY-OPERATING CONDITIONS

Thrust Load:	**13,345 newtons per bearing; 3000 lbs per bearing**	
Temperature:	**Oil-in 366.5°K ±1.7°K; 200°F ± 3°F**	
Oil Flow Rate:	**121 x 10^{-3} kg/sec/bearing;**	**16 lbs/min/bearing**
	106	**14**
	90.7	**12**
	75.6	**10**
	45.4	**6**
Speed:	**8,000 rpm**	**(1.0 x 10^6 DN)**
	16,000	**(2.0 x 10^6 DN)**
	19,200	**(2.4 x 10^6 DN)**
	22,400	**(2.8 x 10^6 DN)**
	24,000	**(3.0 x 10^6 DN)**

In the previous bearing-performance evaluations, the bearing thrust load was 11,121 newtons (2500 lbs) per bearing. The thrust load was increased in Task II to 13,345 newtons (3000 lbs) to ensure sufficient bearing loading at all oil flow rates to prevent destructive skidding at 3.0 million DN. An oil supply nominal temperature of 366.5°K (200°F) was selected for these tests to approximate the temperature expected in engine operation. This oil supply temperature had been used in tests of the bearings under Contract NAS 3-13491. Since bearing hardware can experience sudden changes in temperature at high speeds, a maximum bearing outer-ring temperature was established at 491.5°K (425°F) to prevent damage from excessive temperature. Bearing lubricant was supplied at five different flow rates on the basis of an oil-transfer-scoop nominal efficiency of 76% ± 3% at most operating conditions as determined in Task I.

TEST PROCEDURE

Testing was started with two solid-ball bearings (S/N 2528 A-1 and S/N 2528 A-2), and the entire procedure was repeated with two drilled-ball bearings (S/N 2552 A-1 and S/N 2552 A-2) to obtain comparative test data. Testing was intiated at 1.0 million DN and progressed to successively higher DN values. At each DN value above 1.0 million, oil flow was set initially at 121 x 10^{-3} kilograms per second (16 lbs/min) and then reduced to lower flows, as indicated in Table VI. At 1.0 million DN, the maximum bearing oil supply was 95.8 kilograms per second (12.67 lbs/min) because of decreased scoop efficiency discovered during Task I testing. Bearing operating conditions were maintained constant at each oil flow rate while thermal stability was being established. Thermal stability was assumed to exist when the difference between outer-race temperature and oil-in temperature did not change by more than 1.1°K (2°F) in a period of five minutes. Testing at any combination of oil flow-rate and DN value was to be abandoned if thermal stability could not be achieved within one hour or if bearing outer-ring temperatures exceeded 491.5°K (425°F). Testing of any bearing pair was to be terminated if a bearing failure occurred at any test condition.

Testing performed under Contract NAS3-13491 showed that the specific drilled-ball bearing design was slightly sensitive to startup and shutdown operating conditions. Minor surface distress could be induced by thrust-loading the drilled-ball bearings at zero speed; therefore, a startup-shutdown procedure, originally developed under Contract NAS3-13491, was used with the two drilled-ball bearings to eliminate this problem in the Oil Flow Rate Tests of Task II. Details of the procedure are presented in Table VII.

TABLE VII
STARTUP-SHUTDOWN PROCEDURES FOR DRILLED BALL BEARINGS

STARTUP

1. Set rig shaft speed at 1000 rpm
2. Increase bearing thrust load to 1,112 - 1,668 newtons (250-375 lbs)
3. Increase rig speed to 6000 rpm
4. Increase thrust load to 4,448 newtons (1000 lbs)
5. Increase rig speed to 8000 rpm
6. Increase thrust load to 13,345 newtons (3000 lbs)
7. Increase rig speed to test condition

SHUTDOWN

1. Decrease rig shaft speed to 8000 rpm
2. Decrease bearing thrust load to 4,448 newtons (1000 lbs)
3. Decrease rig speed to 6000 rpm
4. Decrease thrust load to 2,224 newtons (500 lbs)
5. Decrease rig speed to 4000 rpm
6. Decrease thrust load to 1,112 - 1,668 newtons (250-375 lbs)
7. Decrease rig speed to 1000 rpm
8. Decrease rig speed and thrust load simultaneously to zero

SOLID-BALL BEARING PERFORMANCE

Temperature and cage speed data for the two solid-ball bearings (S/N 2528A-1 and S/N 2528A-2) are summarized in Tables VIII and IX.

The oil-inlet temperature was held at 366.5°K ±1.7°K (200°F ±3°F) over the range of shaft speeds and bearing oil-supply flows. The two bearings demonstrated almost identical performance up to 24,000 rpm (3.0 million DN) at all lubricant flow rates. At each shaft speed. the outer-race and oil-outlet average temperatures increased almost linearly with decreases in bearing lubricant from the maximum flow rate down to 75.6 x 10^{-3} kilograms per second (10 lbs/min). However, further lubricant reduction to the minimum flow of 45.4 x 10^{-3} kilograms per second (6 lbs/min) increased the rate of temperature rise substantially. The front bearing ran slightly warmer than the rear bearing, ranging from 1.1°K (2°F) at 8,000 rpm to 3.3°K - 8.9°K (6°F - 16°F) at 24,000 rpm for all oil flows. Increasing the shaft speed produced corresponding increases in outer-race and oil-outlet average temperatures over the range of oil flows. At 3.0 million DN the minimum outer-race and oil-outlet average temperatures were produced at the maximum bearing lubricant flow of 121 x 10^{-3} kilograms per second (16 lbs/min).

The cage speed measurements of the two solid-ball bearings did not indicate any problem with destructive ball skidding up to 3.0 million DN (24,000 rpm). Bearing cage speeds ranged between 44.0 and 45.7% of shaft speed for most combinations of shaft speed and oil flow up to 22,400 rpm (2.8 million DN). At 24,000 rpm, cage speeds decreased slightly, ranging between 43.4 and 43.9% of shaft speed.

A tabulation of the thermal stability data for the solid-ball bearing performance calibration is included in columns 15 and 16 of Tables VIII and IX. It is readily apparent that the difference between the outer-race and oil-in temperatures (ΔT) changed 1.1°K (2°F) or less during a five minute period of stable operation at each set of operating conditions. Thermal stability was usually achieved within 10 to 30 minutes after setting each test point.

The two solid-ball bearings accumulated 23.15 hours running time during the Task II tests. The running time at each set of operating conditions is presented in Tables VIII and IX.

DRILLED-BALL BEARING PERFORMANCE

Temperature and cage speed data for the two drilled-ball bearings (S/N 2552A-1 and S/N 2552A-2) are summarized in Tables X and XI.

The performances of both drilled-ball bearings were identical at all combinations of speed and oil flow except at 24,000 rpm with an oil flow of 45.4 x 10^{-3} kilograms per second (6 lbs/min) per bearing. Testing was terminated after 50 minutes at this final test point because the outer race temperature of the front bearing (S/N 2552A-1) suddenly increased from 490.93°K (425°F) to 501.48°K (443°F) within one minute. Thermal stability was attained at all shaft speeds and bearing lubricant flow rates, usually within 10 to 30 minutes after setting each test point.

TABLE VIII

SOLD-BALL BEARING PERFORMANCE

(1) Thrust Load (newtons)	(2) DN x 10^6	(3) Shaft Speed (rpm)	NO. 2 BEARING (REAR) P/N SKN52575 S/N 2528A-2 (4) Cage Speed Percent*	(5) Oil Inlet Temp (°K)	(6) Avg. Outer Race Temp (°K)	(7) Outer Race Oil Inlet ΔT (°K)	(8) Avg. Oil Outlet Temp (°K)	(9) Bearing Oil Flow Rate (10^{-3}kg/sec)	NO. 1 BEARING (FRONT) P/N SKN 52575 S/N2528A-1 (10) Cage Speed Percent*	(11) Oil Inlet Temp (°K)	(12) Avg. Outer Race Temp (°K)	(13) Outer Race Oil Inlet ΔT (°K)	(14) Avg. Oil Outlet Temp (°K)	(15) Outer Race - Oil Inlet Measured ΔT (°K)**	(16) ΔT Variation For A 5 Minute Period	(17) Bearing Oil Flow Rate (10^{-3} kg/sec)	(18) Running Time (hrs)
13345	1.0	8,000	44.85	367.59	383.71	16.12	379.26	92.2	44.85	367.59	384.26	16.67	379.82	17.09-17.26	(0.17)	92.2	1.00
13345	1.0	8,000	44.92	366.48	383.15	16.67	378.71	94.5	44.92	366.48	383.15	16.67	379.26	16.89-17.00	(0.11)	94.5	.75
13345	1.0	8,000	44.85	365.93	383.15	17.22	378.71	90.7	44.85	365.93	383.15	17.22	379.26	17.78-1787	(0.09)	90.7	.67
13345	1.0	8,000	44.85	366.48	384.26	17.78	380.37	75.6	44.82	366.48	385.37	18.89	381.48	19.83-20.16	(0.33)	75.6	.75
13345	1.0	8,000	44.85	367.59	394.82	27.23	388.15	45.4	44.78	367.59	394.26	26.67	389.26	25.33-26.05	(0.72)	45.4	1.00
13345	2.0	16,000	44.39	365.37	405.93	40.56	400.93	121.0	45.01	365.37	408.71	43.34	404.82	44.50-44.94	(0.44)	121.0	.92
13345	2.0	16.000	45.37	367.59	410.93	43.34	404.82	106.0	44.98	367.59	413.15	45.56	408.71	46.77-47.16	(0.39)	106.0	.75
13345	2.0	16,000	45.40	366.48	413.71	47.23	407.04	90.7	44.99	366.48	415.93	49.45	412.04	49.83-50.38	(0.55)	90.7	.84
13345	2.0	16,000	45.24	366.48	416.48	50.00	408.71	75.6	44.83	366.48	420.37	53.89	415.93	54.61-54.72	(0.11)	75.6	.75
13345	2.0	16,000	45.37	366.48	433.15	66.67	420.93	45.4	44.89	366.48	434.82	68.34	427.04	67.33-68.44	(1.11)	45.4	1.00
13345	2.4	19,200	44.98	368.15	421.48	53.33	415.93	121.0	44.53	368.15	424.82	56.67	419.82	57.00-57.33	(0.33)	121.0	.75
13345	2.4	19,200	45.01	366.48	423.15	56.67	417.04	106.0	44.56	366.48	425.93	59.45	421.48	60.55-60.72	(0.17)	106.0	.83
13345	2.4	19,200	45.04	368.15	427.59	59.44	420.37	90.7	44.65	368.15	432.04	63.89	425.37	65.05-65.66	(0.61)	90.7	1.25
13345	2.4	19,200	44.80	365.93	428.71	62.78	422.59	75.6	44.49	365.93	434.26	68.33	429.82	68.10-68.66	(0.56)	75.6	.67
13345	2.4	19,200	47.99	366.48	447.59	81.11	437.59	45.4	45.88	366.48	450.37	83.89	440.93	84.10-85.21	(1.11)	45.4	1.00
13345	2.8	22,400	45.64	366.48	431.48	65.00	427.04	121.0	44.66	366.48	433.71	67.23	429.82	68.88-69.44	(0.56)	121.0	.84
13345	2.8	22,400	44.16	367.04	433.15	66.11	428.15	106.0	43.90	367.04	437.04	70.00	433.15	70.94-71.33	(0.39)	106.0	.67
13345	2.8	22,400	44.35	366.48	436.48	70.00	430.93	90.7	44.02	366.48	440.93	74.45	436.48	74.05-74.44	(0.39)	90.7	1.17
13345	2.8	22,400	44.52	365.93	443.15	77.22	437.59	75.6	44.23	365.93	449.82	83.89	444.26	84.32-84.49	(0.17)	75.6	.67
13345	2.8	22,400	44.14	366.48	463.71	97.23	453.71	45.4	44.11	366.48	470.37	103.89	455.93	102.55-103.60	(1.05)	45.4	1.00
13345	3.0	24,000	43.49	368.15	437.59	69.44	432.59	121.0	43.43	368.15	440.93	72.78	438.15	73.27-73.66	(0.39)	121.0	1.00
13345	3.0	24,000	43.63	365.37	438.15	72.78	432.59	106.0	43.47	365.37	442.04	76.67	438.71	76.77-77.05	(0.28)	106.0	.66
13345	3.0	24,000	43.84	366.48	442.59	76.11	437.04	90.7	43.63	366.48	448.15	81.67	443.71	81.66-82.05	(0.39)	90.7	1.00
13345	3.0	24,000	43.81	364.82	448.15	83.33	442.04	75.6	43.58	364.82	457.04	92.22	450.37	91.94-92.44	(0.50)	75.6	1.00
13345	3.0	24,000	43.90	366.48	479.82	113.34	464.26	45.4	43.58	366.48	484.82	118.34	465.37	117.99-118.38	(0.39)	45.4	1.00

* Cage Speed is Expressed as a Percent of Shaft RPM.

** Bearing Outer Race - Oil Inlet ΔT is Measured Directly by a Supplementary Thermocouple Circuit for Determining Thermal Stability.

TABLE IX
SOLID-BALL BEARING PERFORMANCE

			NO. 2 BEARING (REAR) P/N SKN 52575 S/N 2528A-2						NO. 1 BEARING (FRONT) P/N SKN 52575 S/N 2528A-1								
(1) Thrust Load (lbs)	(2) DN x 10^6	(3) Shaft Speed (rpm)	(4) Cage Speed Percent*	(5) Oil Inlet Temp (°F)	(6) Avg. Outer Race Temp (°F)	(7) Outer Race Oil Inlet ΔT (°F)	(8) Avg. Oil Outlet Temp (°F)	(9) Bearing Oil Flow Rate (PPM)	(10) Cage Speed Percent*	(11) Oil Inlet Temp. (°F)	(12) Avg. Outer Race Temp. (°F)	(13) Outer Race Oil Inlet ΔT (°F)	(14) Avg. Oil Outlet Temp (°F)	(15) Outer Race - Oil Inlet Measured ΔT (°F)**	(16) ΔT Variation For A 5 Minute Period	(17) Bearing Oil Flow Rate (PPM)	(18) Running Time (hrs)
3,000	1.0	8,000	44.85	202	231	29	223	12.20	44.85	202	232	30	224	30.8 - 31.1	(0.3)	12.20	1.00
3,000	1.0	8,000	44.92	200	230	30	222	12.50	44.92	200	230	30	223	30.4 - 30.6	(0.2)	12.50	.75
3,000	1.0	8,000	44.85	199	230	31	222	12	44.85	199	230	31	223	32.0 - 32.2	(0.2)	12	.67
3,000	1.0	8,000	44.85	200	232	32	225	10	44.82	200	234	34	227	35.7 - 36.3	(0.6)	10	.75
3,000	1.0	8,000	44.85	202	251	49	239	6	44.78	202	250	48	241	45.6 - 46.9	(1.3)	6	1.00
3,000	2.0	16,000	44.39	198	271	73	262	16	45.01	198	276	78	269	80.1 - 80.9	(0.8)	16	.92
3,000	2.0	16,000	45.37	202	280	78	269	14	44.95	202	284	82	276	84.2 - 84.9	(0.7)	14	.75
3,000	2.0	16,000	45.40	200	285	85	273	12	44.99	200	289	89	282	89.7 - 90.7	(1.0)	12	.84
3,000	2.0	16,000	45.24	200	290	90	276	10	44.83	200	297	97	289	98.3 - 98.5	(0.2)	10	.75
3,000	2.0	16,000	45.37	200	320	120	298	6	44.89	200	323	123	309	121.2 - 123.2	(2.0)	6	1.00
3,000	2.4	19,200	44.98	203	299	96	289	16	44.53	203	305	102	296	102.6 - 103.2	(0.6)	16	.75
3,000	2.4	19,200	45.01	200	302	102	291	14	44.56	200	307	107	299	109.0 - 109.3	(0.5)	14	.83
3,000	2.4	19,200	45.04	203	310	107	297	12	44.65	203	318	115	306	117.1 - 118.3	(1.2)	12	1.25
3,000	2.4	19,200	44.80	199	312	113	301	10	44.49	199	322	123	314	122.6 - 123.6	(1.0)	10	.67
3,000	2.4	19,200	47.99	200	346	146	328	6	45.88	200	351	151	334	151.4 - 153.4	(2.0)	6	1.00
3,000	2.8	22,400	45.64	200	317	117	309	16	44.66	200	321	121	314	124.0-125.0	(1.0)	16	.84
3,000	2.8	22,400	44.16	201	320	119	311	14	43.90	201	327	126	320	127.7 - 128.4	(0.7)	14	.67
3,000	2.8	22,400	44.35	200	326	126	316	12	44.02	200	334	134	326	133.3 - 134.0	(0.7)	12	1.17
3,000	2.8	22,400	44.52	199	338	139	328	10	44.23	199	350	151	340	151.8 - 152.1	(0.3)	10	.67
3,000	2.8	22,400	44.14	200	375	175	357	6	44.11	200	387	187	361	184.6 - 186.5	(1.9)	6	1.00
3,000	3.0	24,000	43.49	203	328	125	319	16	43.43	203	334	131	329	131.9 - 132.6	(0.7)	16	1.00
3,000	3.0	24,000	43.63	198	329	131	319	14	43.47	198	336	138	330	138.2 - 138.7	(0.5)	14	.66
3,000	3.0	24,000	43.84	200	337	137	327	12	43.63	200	347	147	339	147.0 - 147.7	(0.7)	12	1.00
3,000	3.0	24,000	43.81	197	347	150	336	10	43.58	197	363	166	351	165.5 - 166.4	(0.9)	10	1.00
3,000	3.0	24,000	43.90	200	404	204	376	6	43.58	200	413	213	378	212.4 - 213.1	(0.7)	6	1.00

* Cage Speed is Expressed as a Percent of Shaft rpm.

** Bearing Outer Race - Oil Inlet ΔT is Measured Directly by a Supplementary Thermocouple Circuit for Determining Thermal Stability.

TABLE X
DRILLED-BALL BEARING PERFORMANCE

(1) Thrust Load (newtons)	(2) DN x 10^6	(3) Shaft Speed (rpm)	NO. 2 BEARING (REAR) P/N SKN 52576 S/N 2552A-2: (4) Cage Speed Percent*	(5) Oil Inlet Temp (°K)	(6) Avg. Outer Race Temp (°K)	(7) Outer Race Oil Inlet ΔT (°K)	(8) Avg. Oil Outlet Temp (°K)	(9) Bearing Oil Flow Rate (10^{-3} kg/sec)	NO. 1 BEARING (FRONT) P/N SKN 52576 S/N 2552A-1: (10) Cage Speed Percent*	(11) Oil Inlet Temp (°K)	(12) Avg. Outer Race Temp (°K)	(13) Outer Race Oil Inlet ΔT (°K)	(14) Avg. Oil Outlet Temp (°K)	(15) Outer Race - Oil Inlet Measured ΔT (°K)**	(16) ΔT Variation For A 5 Minute Period	(17) Bearing Oil Flow Rate (10^{-3} kg/sec)	(18) Running Time (hrs)
13345	1.0	8,000	44.32	364.82	379.26	14.44	374.82	92.2	44.32	364.82	379.26	14.44	376.48	15.50-15.61	(0.11)	92.2	1.00
13345	1.0	8,000	44.53	366.48	380.93	14.45	376.48	94.5	44.53	366.48	380.93	14.45	377.59	14.83-15.00	(0.17)	94.5	.75
13345	1.0	8,000	44.21	366.48	381.48	15.00	376.48	90.7	44.17	366.48	381.48	15.00	378.15	15.17-15.78	(0.61)	90.7	1.00
13345	1.0	8,000	44.25	365.93	383.15	17.22	377.59	75.6	44.25	365.93	382.59	16.66	379.26	17.00-17.17	(0.17)	75.6	.75
13345	1.0	8,000	44.39	366.48	389.82	23.34	383.15	45.4	44.35	366.48	387.59	21.11	384.26	21.66-21.89	(0.23)	45.4	1.00
13345	2.0	16,000	44.35	367.59	403.15	35.56	397.04	121.0	44.17	367.59	402.59	35.00	399.26	36.16-36.61	(0.45)	121.0	1.00
13345	2.0	16,000	44.62	368.15	406.48	38.33	399.82	106.0	44.35	368.15	405.37	37.22	402.04	38.61-38.89	(0.28)	106.0	1.00
13345	2.0	16,000	44.44	365.93	406.48	40.55	399.82	90.7	44.19	365.93	406.48	40.55	403.71	41.44-41.66	(0.22)	90.7	.75
13345	2.0	16,000	44.67	366.48	413.15	46.67	405.37	75.6	44.42	366.48	413.71	47.23	409.82	48.17-48.44	(0.27)	75.6	.75
13345	2.0	16,000	44.50	366.48	427.04	60.56	417.04	45.4	44.28	366.48	427.04	60.56	420.93	62.10-63.16	(1.06)	45.4	1.00
13345	2.4	19,200	44.22	367.04	415.37	48.33	409.82	121.0	44.00	367.04	414.82	47.78	412.04	48.72-49.05	(0.33)	121.0	.75
13345	2.4	19,200	44.32	365.93	417.59	51.66	411.48	106.0	44.10	365.93	417.04	51.11	413.71	52.38-52.66	(0.28)	106.0	.75
13345	2.4	19,200	44.29	367.59	422.59	55.00	415.93	90.7	44.09	367.59	423.15	55.56	418.71	56.27-56.44	(0.17)	90.7	.75
13345	2.4	19,200	44.31	367.04	426.48	59.44	420.37	75.6	44.16	367.04	428.71	61.67	424.26	62.38-62.83	(0.45)	75.6	.75
13345	2.4	19,200	44.99	367.59	447.59	80.00	437.59	45.4	44.59	367.59	448.71	81.12	437.59	80.77-81.16	(0.39)	45.4	1.25
13345	2.8	22,400	44.80	367.04	434.82	67.78	427.59	121.0	44.47	367.04	432.59	65.55	428.15	66.49-66.88	(0.39)	121.0	1.00
13345	2.8	22,400	44.85	367.04	438.71	71.67	430.93	106.0	44.49	367.04	436.48	69.44	432.04	69.44-69.66	(0.22)	106.0	1.00
13345	2.8	22,400	44.76	366.48	440.93	74.45	433.15	90.7	44.40	366.48	439.26	72.78	434.26	73.05-73.49	(0.44)	90.7	.75
13345	2.8	22,400	44.91	366.48	445.93	79.45	439.26	75.6	44.58	366.48	445.93	79.45	439.26	79.99-80.27	(0.28)	75.6	.75
13345	2.8	22,400	45.26	365.37	469.82	104.45	457.59	45.4	44.82	365.37	469.82	104.45	455.93	102.71-103.27	(0.56)	45.4	1.00
13345	3.0	24,000	44.31	366.48	440.37	73.89	432.59	121.0	44.04	366.48	439.82	73.34	435.37	72.77-73.77	(1.00)	121.0	1.00
13345	3.0	24,000	44.33	365.37	442.04	76.67	435.93	106.0	44.03	365.37	439.82	74.45	435.37	74.88-75.38	(0.50)	106.0	1.00
13345	3.0	24.000	44.34	365.93	445.93	80.00	440.37	90.7	44.02	365.93	443.15	77.22	439.26	77.33-77.88	(0.55)	90.7	1.00
13345	3.0	24,000	44.41	368.15	451.48	83.33	445.37	75.6	44.14	368.15	453.15	85.00	445.93	84.55-85.05	(0.50)	75.6	1.00
13345	3.0	24,000	44.86	364.82	474.82	110.00	462.59	45.4	44.35	364.82	478.15	113.33	462.04	111.21-112.10	(0.89)	45.4	.83

* Cage Speed is Expressed as a Percent of Shaft rpm.

** Bearing Outer Race Oil Inlet ΔT is Measured Directly by a Supplementary Thermocouple Circuit for Determing Thermal Stability

TABLE XI
DRILLED-BALL BEARING PERFORMANCE

(1) Thrust Load (lbs)	(2) DN x 10^6	(3) Shaft Speed (rpm)	NO. 2 BEARING (REAR) P/N SKN 52576 S/N 2552A-2: (4) Cage Speed Percent*	(5) Oil Inlet Temp (°F)	(6) Avg. Outer Race Temp (°F)	(7) Outer Race Oil Inlet ΔT (°F)	(8) Avg. Oil Outlet Temp (°F)	(9) Bearing Oil Flow Rate (PPM)	NO. 1 BEARING (FRONT) P/N SKN 52576 S/N 2552A-1: (10) Cage Speed Percent*	(11) Oil Inlet Temp. (°F)	(12) Avg. Outer Race Temp (°F)	(13) Outer Race Oil Inlet ΔT (°F)	(14) Avg. Oil Outlet Temp. (°F)	(15) Outer Race - Oil Inlet Measured ΔT (°F)**	(16) ΔT Variation For A 5 Minute Period	(17) Bearing Oil Flow Rate (PPM)	(18) Running Time (hrs)
3000	1.0	8,000	44.32	197	223	26	215	12.20	44.32	197	223	26	218	27.9-28.1	(0.2)	12.20	1.00
3000	1.0	8,000	44.53	200	226	26	218	12.50	44.53	200	226	26	220	26.7-27.0	(0.3)	12.50	.75
3000	1.0	8,000	44.21	200	227	27	218	12	44.17	200	227	27	221	27.3-28.4	(1.1)	12	1.00
3000	1.0	8,000	44.25	199	230	31	220	10	44.25	199	229	30	223	30.6-30.9	(0.3)	10	.75
3000	1.0	8,000	44.39	200	242	42	230	6	44.35	200	238	38	232	39.0-39.4	(0.4)	6	1.00
3000	2.0	16,000	44.35	202	266	64	255	16	44.17	202	265	63	259	65.1-65.9	(0.8)	16	1.00
3000	2.0	16,000	44.62	203	272	69	260	14	44.35	203	270	67	264	69.5-70.0	(0.5)	14	1.00
3000	2.0	16,000	44.44	199	272	73	260	12	44.19	199	272	73	267	74.5-75.0	(0.4)	12	.75
3000	2.0	16,000	44.67	200	284	84	270	10	44.42	200	285	85	278	86.7-87.2	(0.5)	10	.75
3000	2.0	16,000	44.50	200	309	109	291	6	44.28	200	309	109	298	111.8-113.7	(1.9)	6	1.00
3000	2.4	19,200	44.22	201	288	87	278	16	44.0	201	287	86	282	87.7-88.3	(0.6)	16	.75
3000	2.4	19,200	44.32	199	292	93	281	14	44.10	199	291	92	285	94.3-94.8	(0.5)	14	.75
3000	2.4	19,200	44.29	202	301	99	289	12	44.09	202	302	100	294	101.3-101.6	(0.3)	12	.75
3000	2.4	19,200	44.31	201	308	107	297	10	44.16	201	312	111	304	112.3-113.1	(0.8)	10	.75
3000	2.4	19,200	44.99	202	346	144	328	6	44.59	202	348	146	328	145.4-146.1	(0.7)	6	1.25
3000	2.8	22,400	44.80	201	323	122	310	16	44.47	201	319	118	311	119.7-120.4	(0.7)	16	1.00
3000	2.8	22,400	44.85	201	330	129	316	14	44.49	201	326	125	318	125.0-125.4	(0.4)	14	1.00
3000	2.8	22,400	44.76	200	334	134	320	12	44.40	200	331	131	322	131.5-132.3	(0.8)	12	.75
3000	2.8	22,400	44.91	200	343	143	331	10	44.58	200	343	143	331	144.0-144.5	(0.5)	10	.75
3000	2.8	22,400	45.26	198	386	188	364	6	44.82	198	386	188	361	184.9-185.9	(1.0)	6	1.00
3000	3.0	24,000	44.31	200	333	133	319	16	44.04	200	332	132	324	131.0-132.8	(1.8)	16	1.00
3000	3.0	24,000	44.33	198	336	138	325	14	44.03	198	332	134	324	134.8-135.7	(0.9)	14	1.00
3000	3.0	24,000	44.34	199	343	144	333	12	44.02	199	338	139	331	139.2-140.2	(1.0)	12	1.00
3000	3.0	24,000	44.41	203	353	150	342	10	44.14	203	356	153	343	152.2-153.1	(0.9)	10	1.00
3000	3.0	24,000	44.86	197	395	198	373	6	44.35	197	401	204	372	200.2-201.8	(1.6)	6	.83

* Cage Speed is expressed as a Percent of Shaft rpm.

** Bearing Outer Race-Oil Inlet ΔT is Measured Directly By a Supplementary Thermocouple Circuit For Determining Thermal Stability.

The oil inlet temperature was held at 366.5°K ± 1.7°K (200°F ±3°F) over the range of shaft speeds and bearing oil-supply flows. At each shaft speed, the outer-race and oil-outlet average temperatures increased almost linearly with decreases in bearing lubricant from the maximum flow rate down to 75.6 x 10^{-3} kilograms per second (10 lbs/min). However a further lubricant reduction to the minimum flow of 45.4 x 10^{-3} kilograms per second (6 lbs/min) increased the rate of temperature rise substantially. Increasing the shaft speed produced corresponding increases in outer-race and oil-outlet average temperatures over the range of oil flows At 3.0 million DN, the minimum outer-race and oil-outlet average temperatures were produced at the maximum bearing lubricant flow of 121 x 10^{-3} kilograms per second (16 lbs/min).

Prior to termination of the drilled-ball bearing tests, thermal stability was attained 42 minutes after the minimum bearing lubricant flow was set at 24,000 rpm. The average outer-race temperatures of the front and rear bearings stabilized at 478.15°K (401°F) and 474.82°K (395°F) respectively. Six minutes later, the average outer-race temperature of the front bearing (S/N 2552A-1) increased to 491.48°K (425°F) after which it remained constant for another two minutes. The temperature of the rear bearing remained essentially constant. At this time, fifty minutes after setting the test point, the average outer-race temperature of the front bearing suddenly increased to 501.48°K (443°F) within one minute. The shaft speed was reduced immediately to 8,000 rpm, and the procedure described in Table VII was employed to reduce shaft speed below 8000 rpm. There was no increase in rig vibration at any time after the bearing exceeded 478.15°K (401°F). Subsequent inspection indicated that a cage rub had occurred on the shoulder lands of both inner rings as discussed in the "Post-Test Inspection of Drilled-Ball Bearings" section.

A comparison of Tables VIII, IX, X, and XI indicates that bearing performances in the drilled-ball tests had been slightly better than that obtained in the solid-ball tests. Until the sudden temperature change of the front bearing, the drilled-ball bearings had operated satisfactorily at all speed levels and lubricant flow rates, with outer-race and oil-outlet temperatures consistently lower than that experienced with the front solid-ball bearing (S/N 2528A-1). The performance of the rear solid-ball bearing (S/N 2528A-2) had been slightly better than the two drilled-ball bearings at 2.8 and 3.0 million DN (22,400 rpm and 24,000 rpm) for most lubricant flow rates.

The cage speed measurements of the two drilled-ball bearings had not indicated any problem with destructive ball skidding throughout the test range of speeds and oil flows. This is identical to the baseline cage-speed experience. Drilled-ball cage speeds ranged between 44.0 and 45.0% of shaft speed for most combinations of shaft speed and oil flows. One definite trend in cage speed values was found to occur in the range of 19,200 rpm (2.4 million DN) and above. Cage speed had increased slightly when the bearing lubricant was reduced to the minimum flow of 45.4 x 10^{-3} kilograms per second (6.0 lbs/min). A similar change had occurred with both solid-ball bearings at 19,200 rpm but not at any other shaft speed. The cage speeds of both drilled-ball bearings (S/N 2552A-1 and S/N 2552A-2) had not exhibited any change after the minimum lubricant flow was established at 24,000 rpm up to termination of testing.

The two drilled-ball bearings accumulated 23.52 hours running time during the Task II tests. The running time at each set of operating conditions is presented in Tables X and XI.

POST-TEST INSPECTION OF SOLID-BALL BEARINGS

The two solid-ball bearings are shown in Figures 16 and 17 in their post test condition. Generally, the inner rings, outer rings, and cages of both bearings were in good condition, and no evidence of ball skidding was present. A number of balls from the front bearing (S/N 2528A-1) contained black surface stains and slight pitting.

Typical ring-surface conditions are shown in Figures 18, 19, and 20. Ball tracks were evident on the outer-race and the load-carrying inner-race contact surfaces. The color of these rings was straw. The nonload carrying inner-race did not contain any ball tracks, and it retained a very faint straw color. The land surfaces on the shoulders of both inner-ring configurations contained light circumferential rubbing contact marks from the cage rails. Light black stain marks were dispersed on the land surfaces of both inner-rings and the raceway of the nonload carrying inner-ring for the front bearing (S/N 2528A-1).

The appearances of typical cage surfaces are shown in Figures 21, 22, and 23. As expected, ball pocket contact had been greater in the cage rotational direction than in the axial direction; however, pocket wear was not excessive. The silver plating in the cage bore along the rail locations of the two bearings was lightly polished through contact with the land surfaces of the inner rings. As shown in Table XII, negligible changes were experienced in the balance of the solid-ball bearing cages.

Typical ball surface conditions are shown in Figures 24, 25, and 26. Balls from both bearings contained orbital markings, or tracks, and had retained a straw color. A number of balls from the front bearing (S/N 2528A-1) also contained black surface stains which were randomly dispersed. Slight pitting had formed near the center of a few stains in several balls as shown in Figure 26. Neither set of solid balls contained any evidence of ball skidding.

TABLE XII
SOLID-BALL AND DRILLED-BALL CAGE UNBALANCE MEASUREMENTS

	Pretest (gm-cm)	Post-Test (gm-cm)
BASELINE RUN:		
2528A-1 (Front)	1.0	1.0
2528A-2 (Rear)	0.5	0.5
DRILLED-BALL RUN:		
2552A-1 (Front)	1.0	31.0
2552A-2 (Rear)	1.5	1.5

POST-TEST INSPECTION OF DRILLED-BALL BEARINGS

The two drilled-ball bearings are shown in Figures 27 and 28 in their post-test condition. Generally, the various components of the rear bearing (S/N 2552 A-2) were in good condition and did not show any evidence of ball skidding. The cage of the front bearing (S/N 2552 A-1) had contacted both inner-ring land surfaces relatively hard, penetrating through the silver-plating into the cage base-material. The cage and all of the drilled balls were intact, and there wasn't any evidence that ball skidding had occurred in the front bearing. Material had been removed from the land surfaces of the two inner-rings.

Rear Bearing (S/N 2552 A-2)

Typical ring-surface conditions of the rear bearing are shown in Figures 29, 30, and 31. Ball tracks were evident on the outer-race and on the load-carrying inner-race contact surfaces. The color of these rings was straw. The nonload carrying inner-race did not contain any ball track and had retained a very faint straw color. The land-surfaces on the shoulders of both inner-ring configurations contained light circumferential rubbing contact marks from the cage rails.

The appearance of the rear bearing cage is shown in Figures 32, 33, and 34. Ball pocket contact had been greater in the cage rotational direction, but pocket wear was not excessive. The silver plating in the cage bore along the rail locations was in excellent condition and contained fewer inner-ring contact marks than the two baseline bearing cages. As shown in Table XII, the change in cage unbalance was negligible. The silver-plating on the ends of approximately six pins was slightly blistered, and the plating on two other pins was cracked and separated from the pin surfaces. Although the plating on the other pins appeared to be in good condition, it was possible to separate the plating from the surface of a number of pins with a sharp knife. All cage pins were still tightly attached to the cage rails, and ball contact marks on the pin circumferential silver-plating were light.

Figure 35 shows the post-test condition of five representative balls out of the twenty-one matched ball lot. The balls contained some light orbital lines, random surface-microscratches, and were straw colored. Neither surface cracks nor oil-sludge deposits were found in the bore of any drilled ball from the rear bearing.

Front Bearing (S/N 2552 A-1)

Figures 36 and 37 show the post-test condition of the front bearing before disassembly. There were no cracked balls; the cage was intact, and all pins were still tightly secured to the cage rails. Material had been removed from the land surface of the two inner-rings, as described in Figure 38, from a relatively hard contact by the cage. The color of both inner-rings ranged from straw in the lower raceway surface to purple and dark-blue along the upper raceway surface adjacent to the worn shoulder-land, indicating a temperature range from 477.59°K (400°F) to 560.93°K (550°F). The color of the outer race was straw. Ball tracks were visible on the outer race and on the load-carrying inner race contact surfaces.

Additional details of the cage from the front bearing are shown in Figures 39, 40, and 41. Contact with both inner-rings had been relatively hard, penetrating through the silver-plating on the cage bore along both rails. Material had been removed in an arc sector of about 3.14 radians (180 degrees) to depths ranging from 0.05 to 0.76 millimeters (2 to 30 mils). As shown in Table XII, cage unbalance had increased substantially after the inner ring contact. Ball-pocket contact had been greater in the cage rotational direction but not excessively. Ball contact marks on the pin circumferential silver-plating generally were not heavy, but the plating had been removed at this contact location from two of the pins. The silver-plating was slightly blistered on the ends of approximately six other pins. Just as with the rear bearing cage, it was possible to separate the plating from the surfaces of a number of pins with a sharp knife.

Figure 42 shows the post-test condition of six representative balls out of the twenty-one matched ball lot. The balls contained some light orbital lines, random surface microscratches, and were straw in color. Thirteen of the twenty-one drilled balls contained various amounts of oil-sludge coating on the bore surface. No cracks were visible in the bore of any ball from the front bearing.

DISCUSSION – TASK II TEST RESULTS AND POST-TEST INSPECTION

The two solid-ball bearings operated successfully to 3.0 million DN in Task II over the range of bearing oil-supply flows investigated. The two drilled-ball bearings performed in a similar manner until one of the bearings suddenly sustained a cage rub at the minimum oil flow of 45.4×10^{-3} kilograms per second (6 lbs/min) per bearing at 3.0 million DN. The cage rub occurred fifty minutes after setting the final test point – only ten minutes before completion of the point and Task II testing.

The cage rub was not related to the drilled-ball concept. It was due to marginal lubrication of the inner-ring land surfaces as a result of insufficient oil flow through the limited number of shoulder oil-passages. Such a rub might have occurred in any of the four bearings tested in Task II at the minimum lubricant flow. Lubrication was poorer at this oil flow and shaft speed. This is evident in the bearing temperature data and the post-test surface appearance of the four bearings. This insufficiency is related partly to the design of the bearing lubricant passages and partly to the orientation of these passages relative to the lubricant passages in the hub assembly.

Lubrication of the test bearings is provided through a combination of passages at the bearing ID as shown in Figure 13. These passages supply lubricant directly to the balls and to the shoulder land-surfaces of the inner rings. The radial oil-slots that lead directly to the inner-race surface are more numerous and larger in size than the inner-ring shoulder passages. Therefore, these slots have the capacity to transport a larger quantity of oil. Each inner ring has several radial slots located between the entrance holes of the shoulder oil-passages. If the inner rings are installed on the hub assembly with the shoulder oil holes out of alignment with the six hub oil-discharge holes, lubricant could be transported preferentially through the slots to the balls rather than through the shoulder passages to the land surfaces. As a result, shoulder land lubrication can become inadequate when bearing lubricant supply is reduced significantly as in Task II.

Shoulder land lubrication can be improved even at low oil-supply flows if the shoulder oil-passages are aligned closer to the six hub oil-discharge holes when the bearing is installed on the hub assembly.

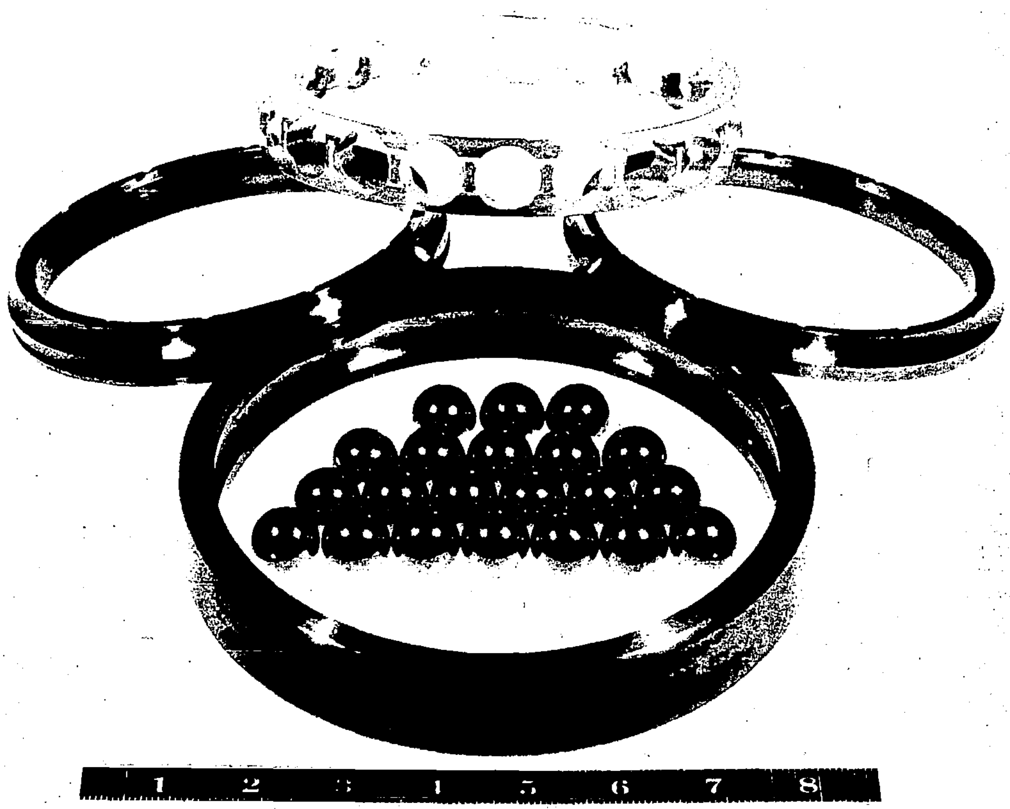


Figure 16   Overall View of Baseline Bearing S/N 2528A-1 After 23.15 Hours

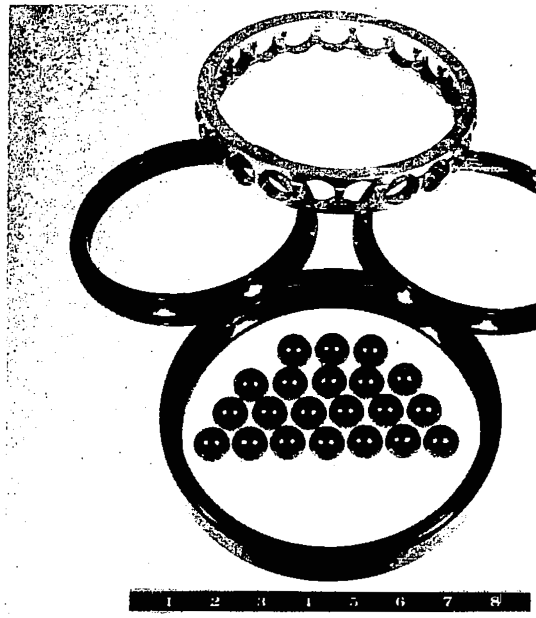


Figure 17 Overall View of Baseline Bearing S/N 2528A-2 After 23.15 Hours

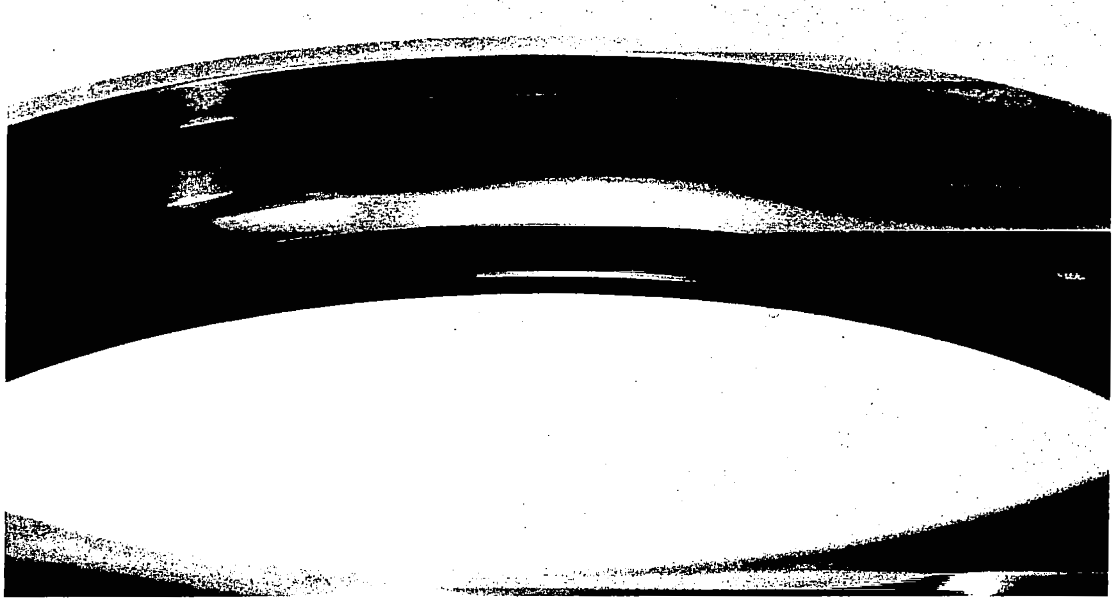


Figure 18 Appearance of Typical Outer-Race – Baseline Bearing S/N 2528A-1

Figure 19 Appearance of Typical Load-Carrying, Split Inner-Ring – Baseline Bearing S/N 2528A-1

Figure 20 Appearance of Typical Unloaded Split Inner-Ring – Baseline Bearing S/N 2528A-1

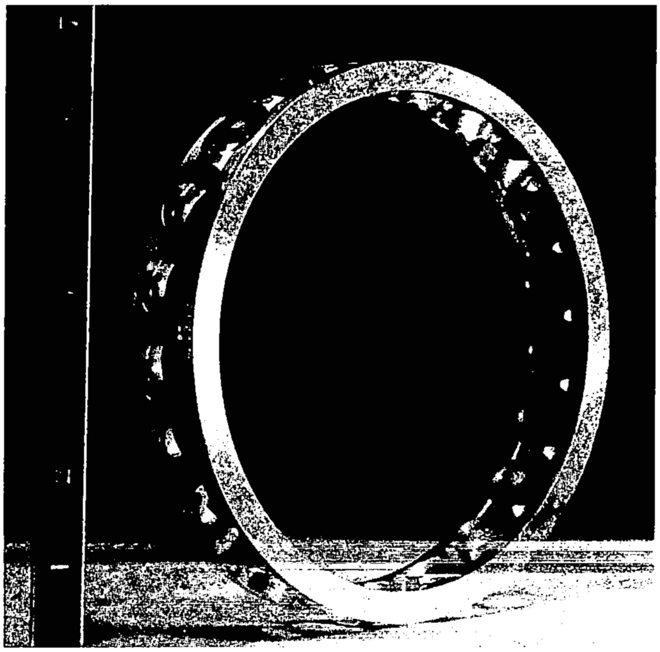


Figure 21 Overall View of Typical Solid-Ball Cage – Baseline Bearing S/N 2528A-1

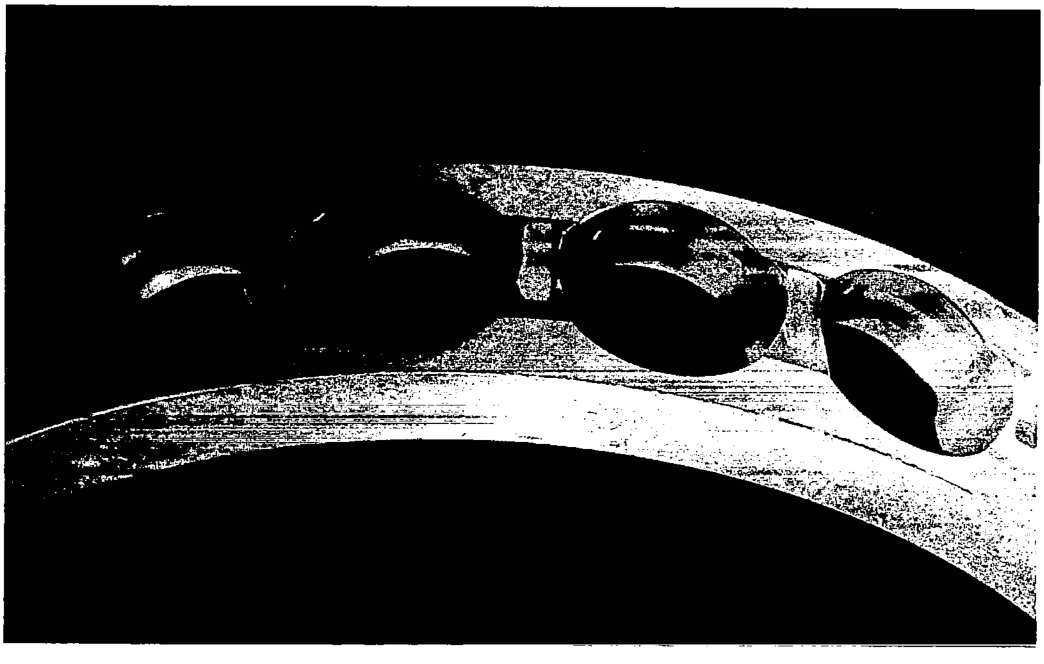


Figure 22 Appearance of Typical Ball Pocket Wear – Baseline Bearing S/N 2528A-1

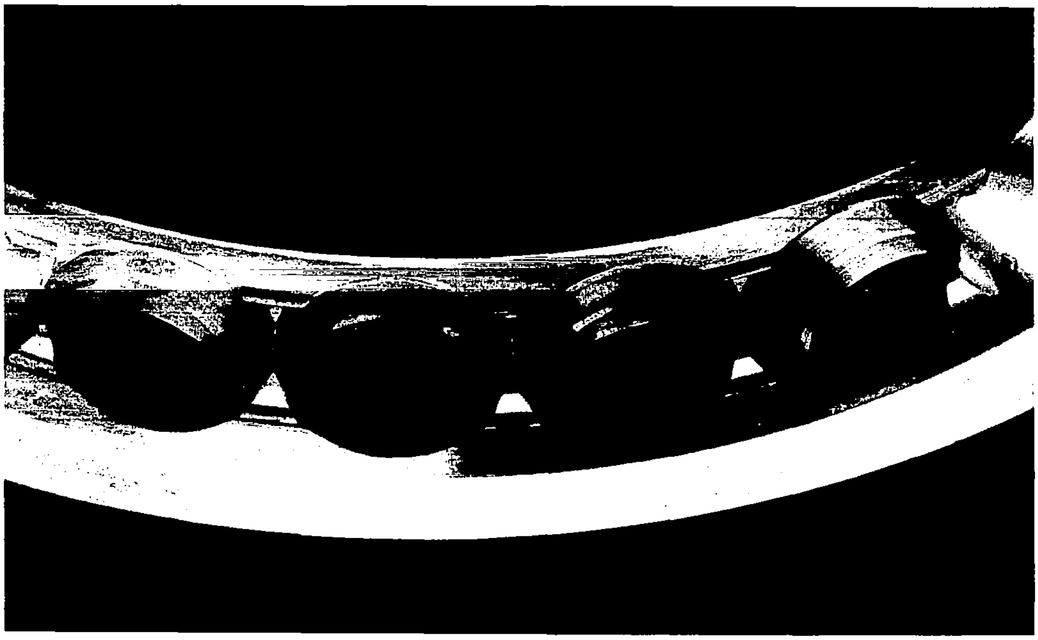


Figure 23   Appearance of Cage Bore and Ball Pocket Wear – Baseline Bearing S/N 2528A-1

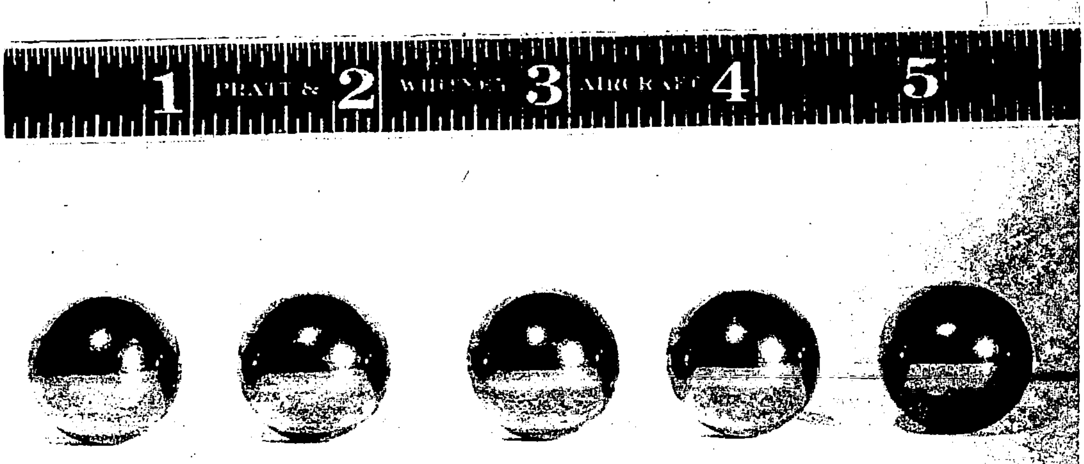


Figure 24   Appearance of Typical Solid Balls – Baseline Bearing S/N 2528A-1

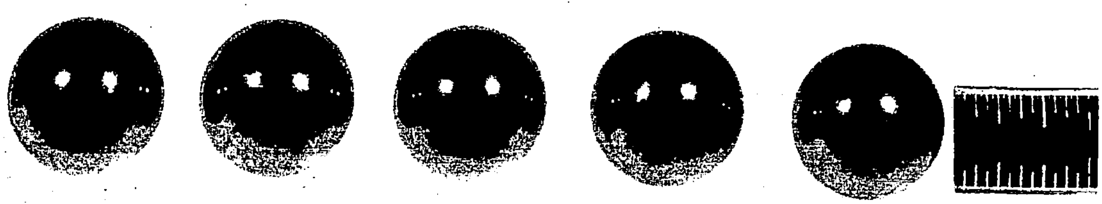


Figure 25 Appearance of Typical Solid Balls – Baseline Bearing S/N 2528A-2

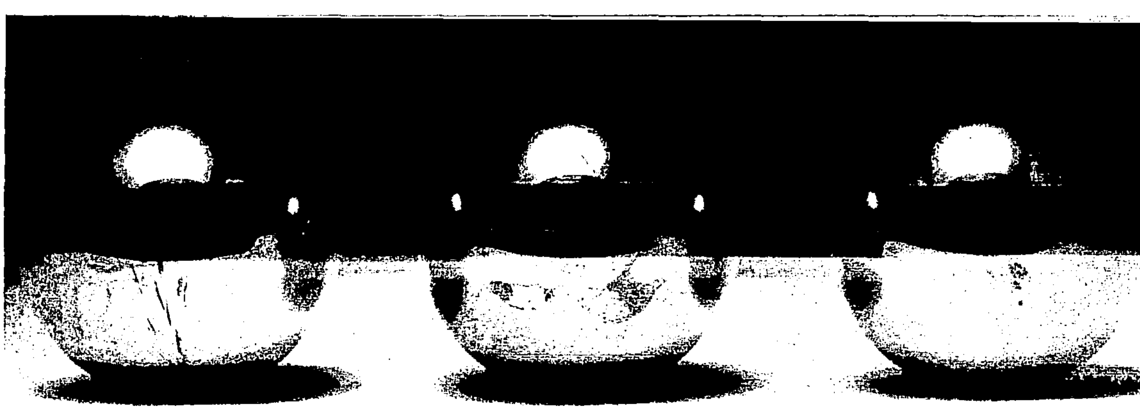


Figure 26 Appearance of Typical Solid Balls With Randomly Dispersed Black Surface Stains and Pitting – Baseline Bearing S/N 2528A-1

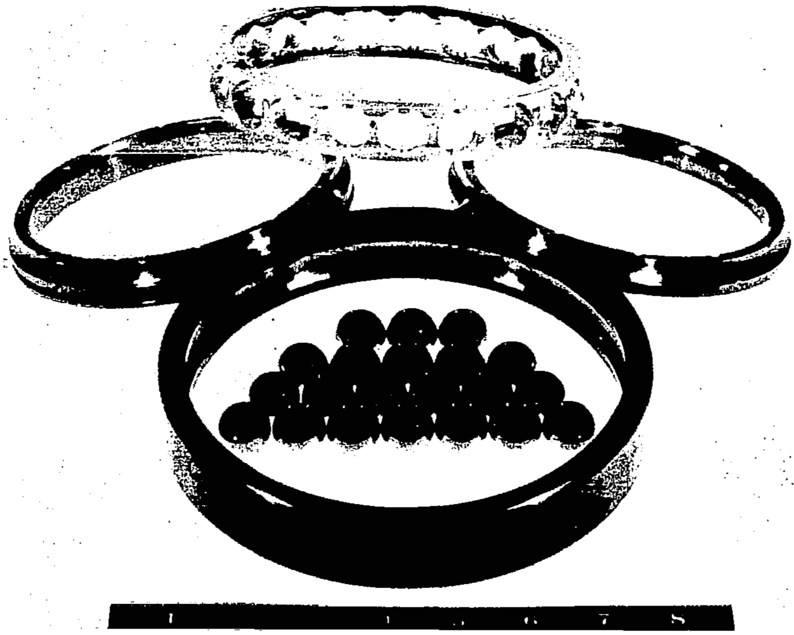


Figure 27 Overall View of Drilled – Ball Bearing S/N 2552A-2 After 23.52 Hours

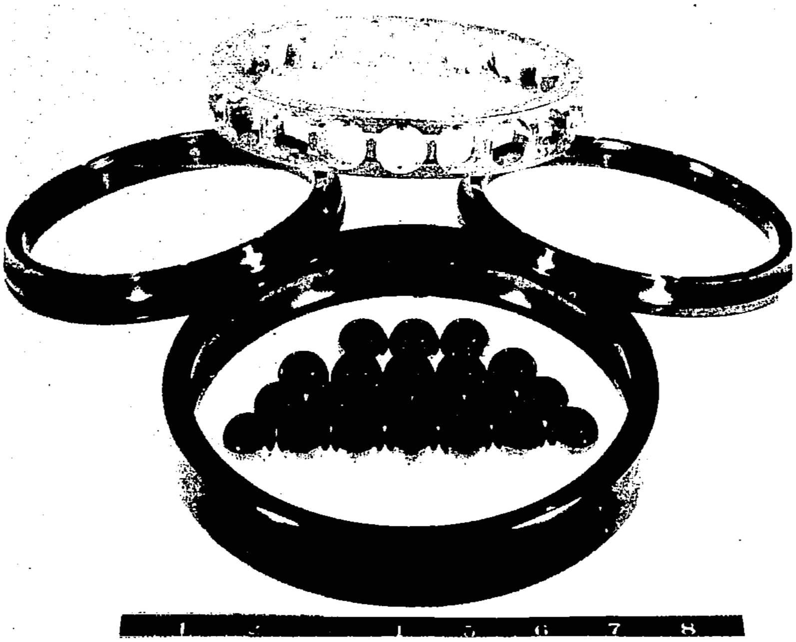


Figure 28 Overall View of Drilled – Ball Bearing S/N 2552A-1 After 23.52 Hours

Figure 29 Appearance of Outer Ring – Drilled-Ball Bearing S/N 2552A-2

Figure 30 Appearance of Thrust-Loaded Split Inner Ring – Drilled-Ball Bearing S/N 2552A-2

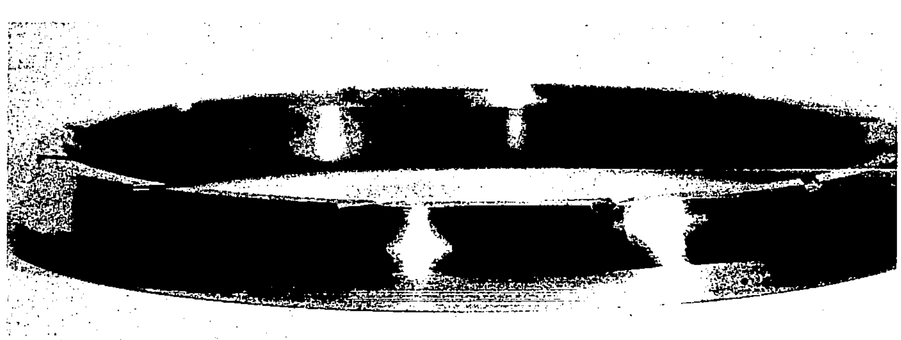


Figure 31 Appearance of Unloaded Split Inner Ring – Drilled-Ball Bearing S/N 2552A-2

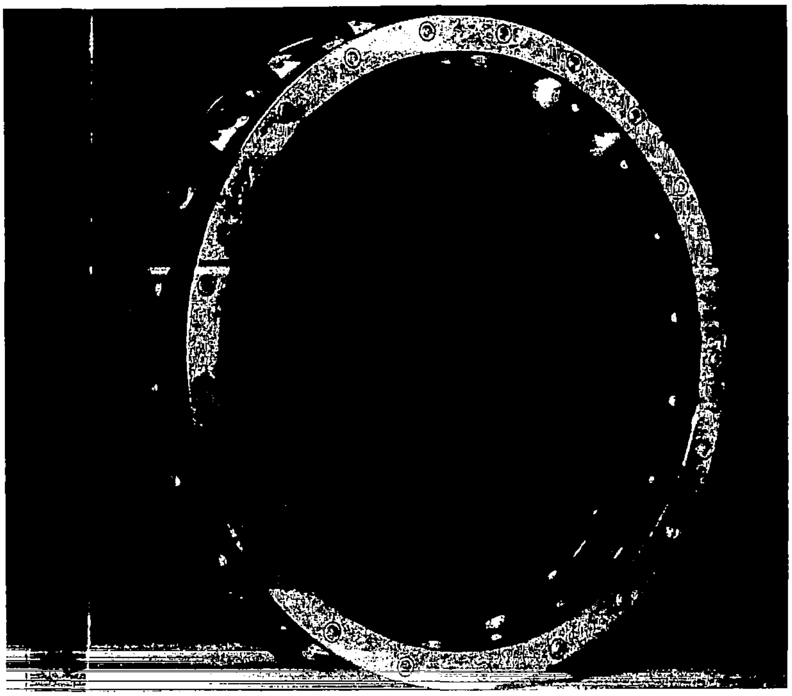


Figure 32 Overall View of Undamaged Drilled-Ball Cage – Drilled-Ball Bearing S/N 2552A-2

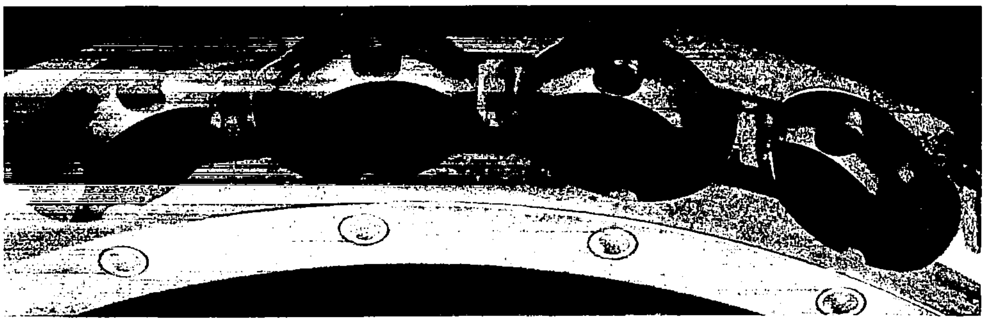


Figure 33 Appearance of Cage Pin and Ball Pocket Wear – Drilled-Ball Bearing S/N 2552A-2

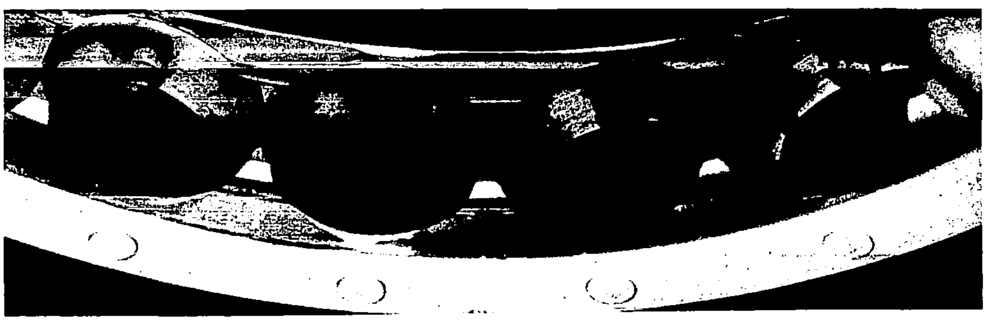


Figure 34 Appearance of Cage Bore, Pin and Ball Pocket Wear – Drilled-Ball Bearing S/N 2552A-2

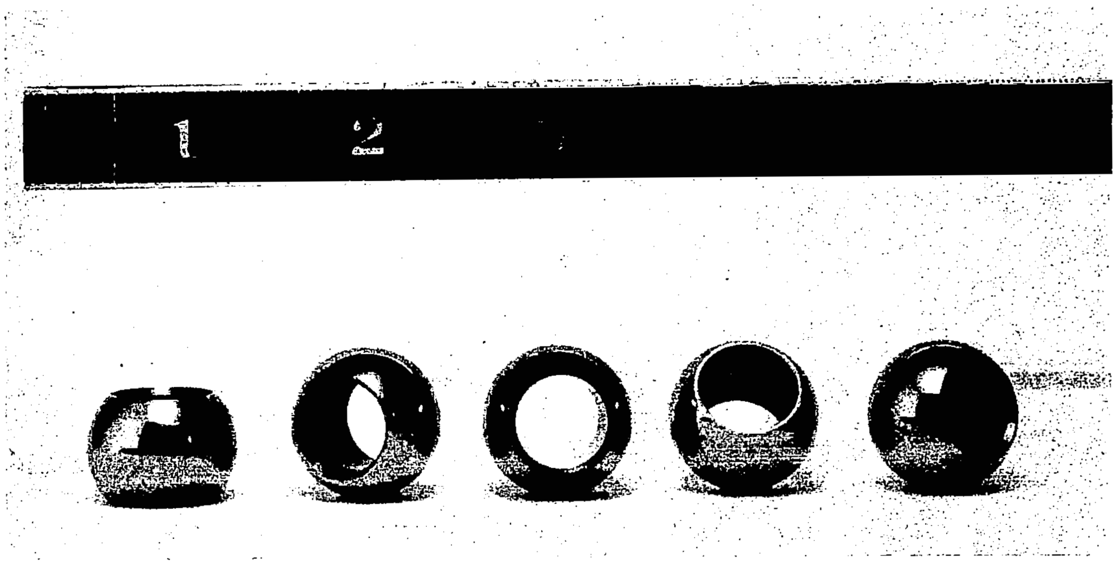


Figure 35 Appearance of Typical Drilled Balls – Bearing S/N 2552A-2

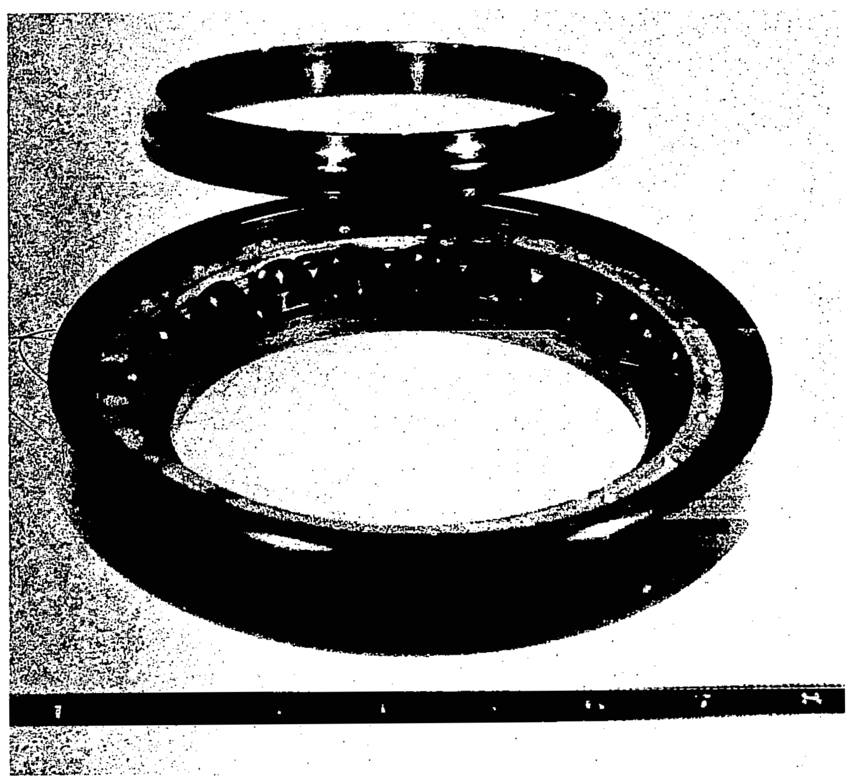


Figure 36 Post-Test Appearance of Drilled-Ball Bearing S/N 2552A-1 Before Disassembly With Thrust-Loaded Inner Ring Removed

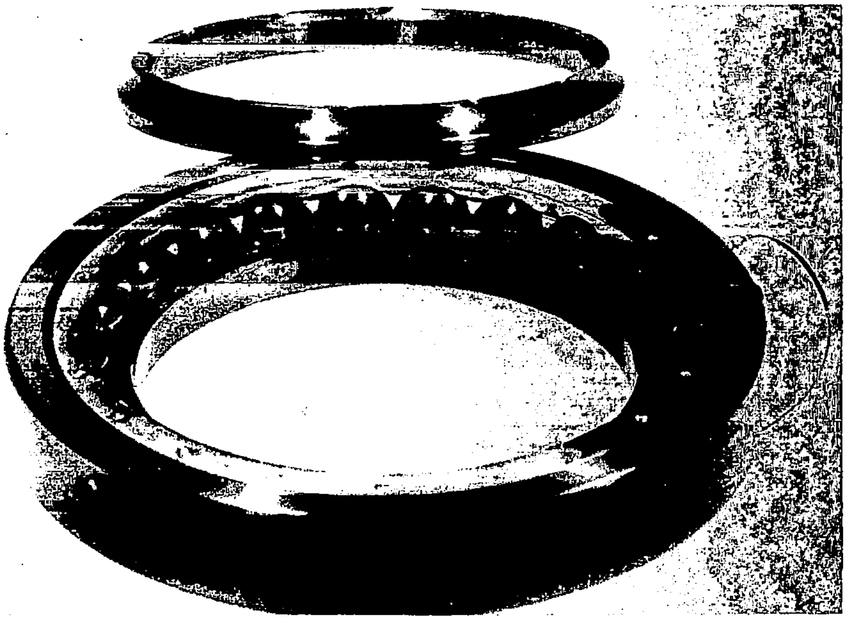


Figure 37 Post-Test Appearance of Drilled-Ball Bearing S/N 2552A-1 Before Disassembly With Unloaded Inner Ring Removed

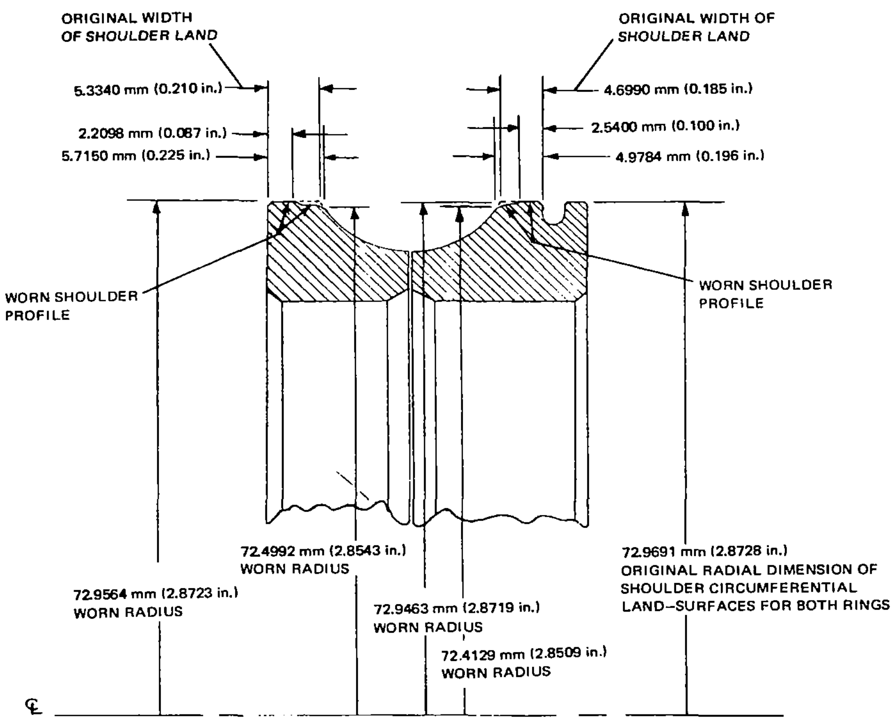


Figure 38 Profile of Worn Inner-Ring Land Surfaces

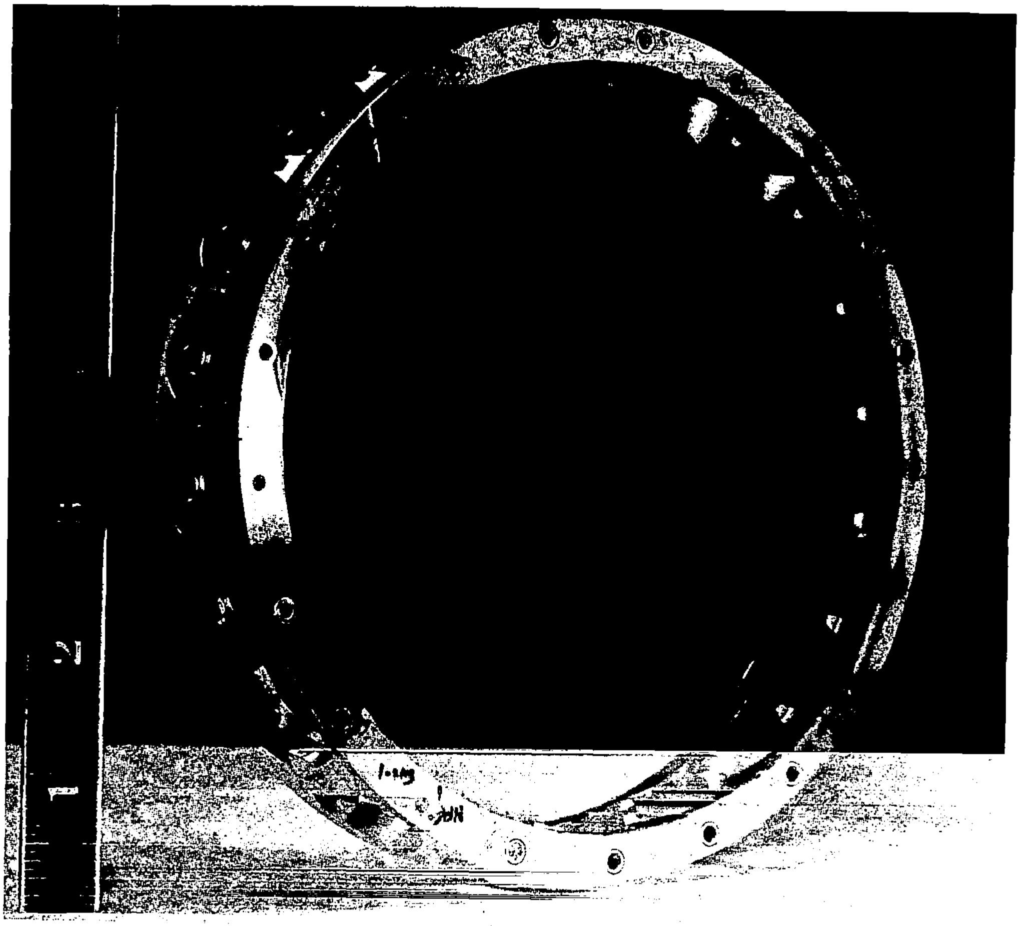


Figure 39 Overall View of Damaged Drilled-Ball Cage – Drilled-Ball Bearing S/N 2552A-1

Figure 40 Appearance of Cage Pin and Ball Pocket Wear – Drilled-Ball Bearing S/N 2552A-1

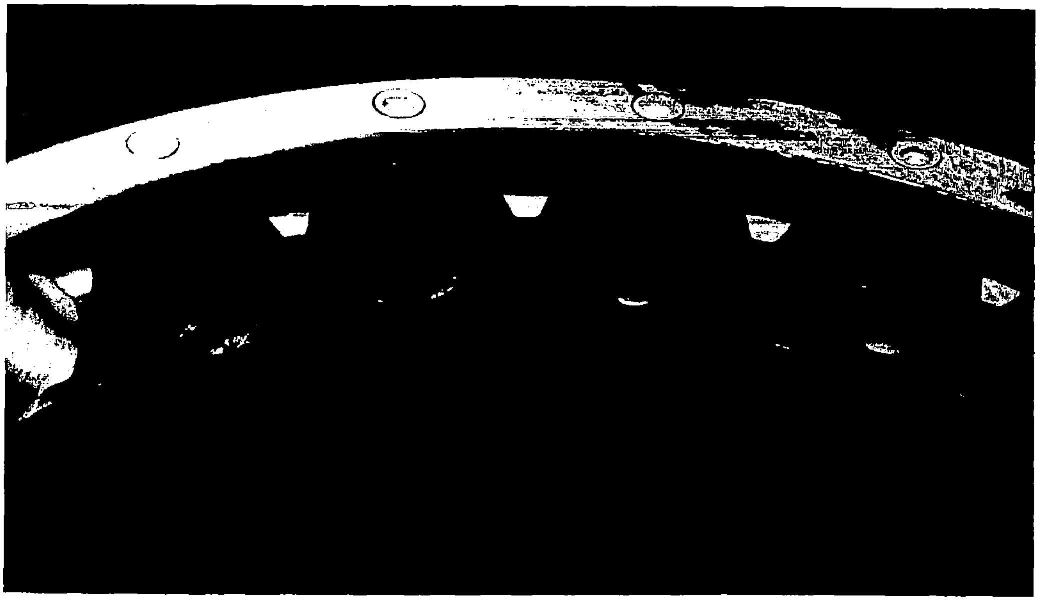


Figure 41 Appearance of Cage Bore, Pin, and Ball Pocket Wear – Drilled-Ball Bearing S/N S/N 2552A-1

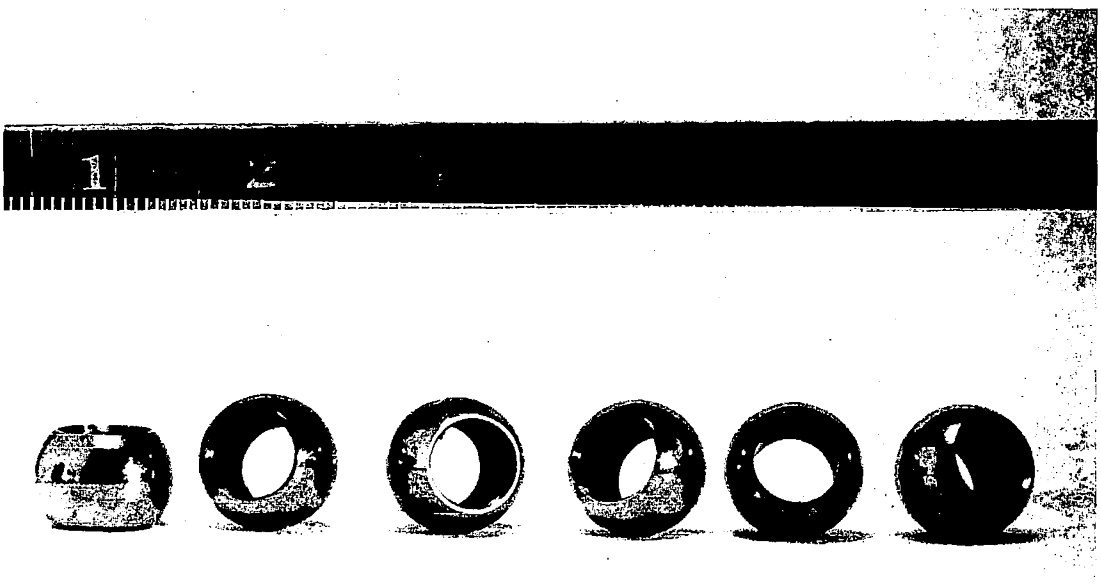


Figure 42 Appearance of Typical Drilled Balls With Oil-Sludge Coating on Bore Surface – Bearing S/N 2552A-1